1984

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BY

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SM (NA&ME)
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Course XIII-A
June 1984

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(1975)

SUBMITTED TO THE DEPARTMENT OF
OCEAN ENGINEERING
IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR THE DEGREES OF

MASTER OF SCIENCE IN NAVAL ARCHITECTURE AND MARINE ENGINEERING

and

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
June, 1984

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Submitted to the Department of Ocean Engineering on May 11, 1984, in partial fulfillment of the requirements for the Degrees of Master of Science in Naval Architecture and Marine Engineering and Master of Science in Mechanical Engineering.

ABSTRACT

A simple mathematical model was developed to simulate the closed portion of the cycle for a quiescent chamber compression ignition engine utilizing the assumption of perfect gases and the first law of thermodynamics. Various input paramentes were used in trend analysis to check the model. The output from the computer program was compared to test data from a four inch bore, open chamber semi-quiescent diesel engine run at the Sloan Test Laboratory. This computer model was then modified to simulate the expansion stroke of a newly developed, two cylinder cycle reciprocating engine. The model was then run to determine the optimum point of fuel injection for the new engine.

Thesis Supervisor: Professor A. Douglas Carmichael

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ACKNOWLEDGEMENTS

I wish to express my sincere gratitude to Professor A. Douglas Carmichael for his understanding, assistance and guidance which made this project a most valuable experience. The discussions with Professors Warren M. Rohsenow and John B. Heywood were also most helpful in keeping me and this project on track.

I wish to thank Mr. Peter Birkholz for the use of his data processing facilities and technical advise during the preparation of the final draft of this thesis.

For their everlasting love and encouragement to do better than they, this project is dedicated to my parents, Wasil and Stella.

Last, but not least by any stretch of the imagination, I am most deeply indebted to my wife, Karen, and my daughters, Kelly and Valery, for their love, patience, understanding and encouragement during the course of this project and for the period of study at M.I.T.

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INTRODUCTION

The compression ignition (CI) engine is thriving in new found popularity amongst automobiles, medium-duty and heavy-duty freight transport trucks, marine propulsion and auxiliary systems, and various other industrial applications. The United States Department of Energy (DOE) has recently predicted that diesel fuel consumption will exceed gasoline consumption in this country by the year 2000. This is primarily due to the shift to diesel power in the automotive and truck freight transport industries to take advantage of the high efficiency, high power-to-weight ratio engines. The present daily diesel fuel consumption of the order of 10^8 liters{1} is expected to increase by as much as 50 percent by the turn of the century. The increasing fuel consumption rate, coupled with the constant concern of diminishing oil reserves, has prompted renewed interest in improving the operating efficiency of the conventional compression ignition engine. Any small improvement in engine efficiency will obviously result in an enormous savings in petroleum.

The approaches currently pursued to improve compression ignition engine efficiency include increasing the compression ratio and the development of the "adiabatic" engine. The former involves turbocharging and the improvement to piston ring technology. The latter approach concentrates the most emphasis on insulating the engine. This requires the use of temperature-resistant ceramic cylinder liners for combustion cylinders whose gas wall temperatures can be of the order of 1200 degrees Kelvin. In addition to these two approaches, there are many other avenues of research in progress that involve improvements that will increase the compression ratio, decrease the heat loss from the engine, or increase the combustion efficiency through improved combustion chamber design.

Instead of improving upon the conventional compression ignition engine, a new cycle engine design is under development. This new design, proposed by Carmichael[2], consists of a two cylinder cycle which divides the functions of a conventional four-stroke diesel cycle into two parts. The new engine has one cylinder which compresses the incoming air charge and another cylinder which acts as the combustion chamber and expansion cylinder. These two cylinders are interconnected by a regenerative heat exchanger. The regenerator acts as the heart of the new design. Through the use of new ceramic materials, the regenerator will act as a heat transfer medium by

transfering a portion of the heat from the exhaust gases to the incoming air charge. The temperature of the incoming air will be elevated twice. The first temperature increase is due to the compression process in the first cylinder. This cylinder, in turn, will transfer its air charge through the ceramic matrix of the regenerative heat exchanger, thus boosting the temperature for the second time. After passing through the regenerator, the incoming air charge will be of sufficient temperature to accomodate spontaneous combustion. With this high temperature, the high compression ratio of the conventional compression ignition engine is not required to obtain work from the cycle. Figures 1 and 2 depict the pressure vs. volume and temperature vs. volume diagrams for the new cycle as compared to a conventional diesel cycle. The preliminary design of this new two cylinder cycle engine indicates that an improvement to thermal efficiency can be achieved over the conventional compression ignition engine.

An important element of the engine design process is the capability to predict, with an acceptable degree of accuracy, the energy release during combustion as a function of time. This process is extremely complex in that it involves the injection and atomization of fuel, the evaporation and mixing of the fuel with the air charge, followed by the various phases of combustion. The ability to accurately predict the heat release rate is vital to the engine designer when analyzing a new engine design.

This thesis is an attempt to assimilate various diesel engine combustion models to produce a simple, yet accurate, model to be used in the continuing evaluation of the new two cylinder cycle reciprocating engine. The proposed model can be utilized on a personal computer system to determine the optimum point of fuel injection for the new engine. The model has the capability to evaluate two different fuel types (i e. iso-octane and propane).

Chapter 2

DESCRIPTION OF THE TWO CYLINDER CYCLE RECIPROCATING ENGINE

The Two Cylinder Cycle Reciprocating Engine consists of one compression cylinder and a pair of combustion/expansion cylinders, see figures 3 and 4. The compression cylinder consists of an intake valve and two exhaust valves, one to each expansion cylinder. Each expansion cylinder has its own fuel injector. The regenerator cavity contains an exhaust valve in addition to the ceramic matrix regenerator. The pistons of both the compression cylinder and the expansion cylinders are considered to be of simple geometry with flat heads. The expansion cylinder piston incorporates no unique features to increase turbulence or swirl, thus it is similar to a direct injection, quiescent chamber diesel engine cylinder. The five valves and three pistons are actuated by a camshaft that allows the compression piston to operate at twice the speed of an expansion piston. The compression cylinder will alternately provide a compressed air charge to each expansion cylinder via the regenerator. A typical cycle can be illustrated by referring to figures 3 and 4.

Step 1: The compression cylinder is at top dead center (TDC) and has just completed transfering an air charge to cylinder A. Cylinder A is just starting an expansion stroke and cylinder B is just starting an exhaust stroke. Valve 1 is closed, valve 2A just closed, valve 3B just opened, valves 2B and 3A are already closed.

Step 2: The compression cylinder is approximately 90^{o} ATDC and in the middle of an air charge induction. Cylinder A is still in the expansion process and cylinder B is still exhausting. Valve 1 is open, valves 2A, 3A, 2B are closed and valve 3B is open.

Step 3: The compression cylinder is at botton dead center (BDC) and has completed induction of an air charge. Cylinder A is still expanding and cylinder B is exhausting through the regenerator. Valve 1 just closed, valves 2A, 3A, 2B are closed and valve 3B is still open.

Step 4: The compression cylinder is approximately 90^{o} BTDC and in the middle of compressing the air charge. Cylinder A is ending its expansion stroke and cylinder B is completing its exhaust stroke. Valve 1 is closed, valves 2A, 3A and 2B are closed, valve 3B is still open.

Step 5: The compression cylinder has just reached TDC and has just completed an impulse air charge transfer to cylinder B through the regenerator. Cylinder A has just commenced its exhaust stroke. Valve 1 is closed, valve 2B

has just closed (it only opened for a very short time just before the compression cylinder reached TDC), valve 3A just opened, valves 2A and 3B are closed. (This is the same as step 1 except that cylinders A and B are reversed.)

<u>Step 6</u>: The compression cylinder is approximately 90° ATDC and is in the middle of an air charge induction. Cylinder A is still exhausting and cylinder B is in the expansion process. Valve 1 is open, valves 2B, 3B, 2A are closed and valve 3A is open. (This is the same as step 2 except that cylinders A and B are reversed.)

<u>Step 7</u>: The compression cylinder has just reached BDC and has completed induction of an air charge. Cylinder A is exhausting through the regenerator and cylinder B is still expanding. Valve 1 just closed, valves 2B, 3B, 2A are closed and valve 3A is open. (This is the same as step 3 except that cylinders A and B are reversed.)

<u>Step 8</u>: The compression stroke is approximately 90° BTDC and in the middle of compressing an air charge. Cylinder A is completing its exhaust stroke and cylinder B is ending its expansion stroke. Valve 1 is closed, valve 2B, 3B and 2A are closed, valve 3A is still open. (This is the same as step 4 except that cylinders A and B are reversed.)

Step 9: This is the same as step 1.

Figures 5 and 6 show the temperature and pressure as a function of cylinder volume for a cycle.

The table below summarizes the sequencing of the valves for a complete cycle of an expansion cylinder.

Table 1: Sequencing of Valves

	Valve				
	1	2A	3A	2B	3B
Step 1:	X	X	X	X	O
Step 2:	O	X	X	X	O
Step 3:	X	X	X	X	O
Step 4:	X	X	X	X	O
Step 5:	X	X	O	X	X
Step 6:	O	X	O	X	X
Step 7:	X	X	O	X	X
Step 8:	X	X	O	X	X
Step 9:	X	X	X	X	O

where X = Valve closed

and O = Valve open

As can be readily seen, the valve timing sequence is rather complex. The timing sequence must be such as to allow the impulse transfer of the air charge to occur

without possible blow-down to the atmosphere or charging the wrong cylinder. A shift of the crank angle must be considered to optimize the air charge transfer sequence to the on-line expansion cylinder. Thus, the valve timing sequence is a critical factor in the correct and efficient operation of this new engine design and must be dealt with appropriately.

Chapter 3

COMBUSTION AND COMBUSTION MODELING

(An overview)

3.1 Description of Diesel Engine Combustion

The diesel engine combustion process is exceedingly complex and not very well understood. Combustion in the diesel engine is characterized by compression ignition, a non-uniform fuel and air distribution in the combustion chamber, and a continuous mixing throughout the period in which combustion occurs. Due to the initial conditions in the chamber when fuel is first injected, the air charge in the cylinder is of sufficient temperature and pressure to support a chain-reaction. However, combustion in the compression ignition engine is governed by the local conditions in each part of the charge and not dependent on the spread of the flame from one point to another. Therefore, the rate of combustion is dependent on the state and distribution of the fuel and upon the pressure and temperature within the cylinder.{3}

3.1.1 The Phases of Combustion

Ricardo described the diesel engine combustion process as taking place in three stages; namely the delay period, a period of rapid combustion, followed by burning at a controlled rate.[3] Lyn[4] described the burning process in three slightly different phases. The first phase is a period of rapid combustion which lasts for only three degrees crank angle. The second stage is characterized by a decreased rate of heat release lasting approximately 40 degrees crank angle. The third period consists of the fuel burning at a very slow rate which may persist through the remainder of the expansion stroke.

A combination of the descriptions of Ricardo and Lyn may be more appropriate. The stages of combustion could be divided into ignition delay, premixed burning, diffusion controlled combustion and the tail of combustion.[5,6] Figure 7 depicts the four stages of combustion in a heat release diagram.

3.1.1.1 Ignition Delay

The term ignition delay, or ignition lag, describes the time required by the preliminary reactions that occur prior to the appearance of flame. The ignition delay is broken down into a physical delay and a chemical delay. The physical delay period occurs between the beginning of fuel injection and the onset of chemical reactions. During this period, the fuel is atomized, vaporized, mixed with air and

raised in temperature. This process is sometimes collectively referred to as preparation. The chemical delay period immediately follows the physical delay period and terminates at inflammation or ignition. This period is characterized by chemical reactions starting slowly with pre-flame oxidation of the fuel followed by local ignition.

The ignition delay will vary according to cylinder temperature, cylinder pressure, the type of fuel, the initial temperature of the fuel, the characteristics of the fuel injectors and the turbulence in the cylinder. The physical delay is small for light fuels but can become the controlling factor for heavy, viscous fuels. The physical delay can be significantly reduced by using high injection pressures and high turbulence to expedite the breakup of the fuel jet.

Semi-empirical relationships have been developed to describe the ignition delay. An estimate for igniton delay was developed by Wolfer in 1938:{7}

$$t = 0.44P^{-1.19}\exp(4650/T)$$

where: t = ignition delay in milliseconds

P = cylinder pressure in atmospheres

and T = temperature in degrees K at ignition.

An estimate by Clarke[8] in 1970 is quite similar to that by Wolfer:

$$t = 0.22\exp(5500/T)P^{-0.727}$$

where: t = ignition delay in seconds

T = cylinder temperature in degrees K

and P = cylinder pressure in N/m^2.

Still another empirical expression for ignition delay was developed by Spadaccini and Tevelde[9] from experiments for NASA in 1979 with diesel fuel in a steady flow facility:

$$t = 2.43 \times 10^{-9} P^{-2} \exp(41560/RT)$$

where t = ignition delay in seconds

P = pressure in atmospheres

T = mixture temperature in degrees K

and R = gas constant in atm $cm^3/gmole^{o}K$.

Figure 8 represents the effects of temperature and pressure on ignition delay as determined from the estimates by Wolfer. The Spadaccini and Tevelde and Clarke relationships yield somewhat similar results.

When using ignition delay expressions, it must be emphasized that differences in engines, fuel properties (especially cetane number), fuel injectors and actual engine temperatures and pressures make the calculation rather approximate. These formulas are also very limited by their use of bulk temperatures, with no consideration of local compositions or temperatures.[10]

3.1.1.2 Premixed Burning

In the premixed burning stage, flame occurs at one or more locations and spreads turbulently. The rate and amount of combustion during this stage is directly related to the fuel preparation rate and the length of the ignition delay period. Since this stage of combustion is one of premixed combustion, little carbon (soot) is produced resulting in little radiation heat transfer. However, since the combustion rate is so intense, combustion generated noise is controlled by this stage of combustion.[11] Figure 9 depicts premixed burning in a cylinder.

3.1.1.3 Diffusion Controlled Burning

Once the prepared, or premixed, fuel has burned, the combustion process slows down. The combustion rate in this stage will be dominated by the rate of local air entrainment. Since the temperature in the cylinder is favorable for ignition in this stage, the air/fuel mixing process will control the rate of combustion. This preparation of the fuel will be governed by the turbulence and swirl in the cylinder. Lyn[4] estimated that approximately 40 percent of the heat release from the combustion of fuel comes from this stage. Figure 10 shows the diffusion burning process in a cylinder.

3.1.1.4 Combustion Tail Stage

This last stage of combustion is characterized by the cylinder pressure and temperature falling as the expansion process continues. The rate of combustion tails off due to the chemical kinetic effects as

the chemical reaction rate slows. In this stage, the reaction rate will become the controlling factor instead of the air/fuel mixing process. This stage is also characterized by diffusion combustion with a high production and combustion of soot particles with a resultant high rate of radiation heat transfer. This last stage of combustion can proceed through the completion of the expansion stroke and can contribute upto 20 percent of the total heat release.[4] Figure 11 represents a typical heat release rate diagram showing the four stages of combustion.

3.2 Combustion Modeling

The combustion process is often considered the most important aspect of an internal combustion engine, but, at the same time, the least understood and most complex. A mathematical model depicting combustion would require good models of the fuel system to include the injection/fuel pump, the injector nozzles, and fuel lines. Additionally, models of fuel atomization, vaporization, fuel/air mixing, cylinder air motion, chemical kinetics and pre-mixed and diffusion mixing would be required. A model as comprehensive as this has yet to be deveoped. Spaulding[12] states that this type of "combustion modeling is impossible." He justifies this by pointing out that the number of governing restraints and rules outnumber the degrees of freedom and, in addition, the requirements of low cost, speed and accuracy must also be met. Since the complexity of the real combustion process is so overwhelming, substantial simplifying assumptions must be made to obtain solutions.

3.2.1 Types of Models and Uses

Bracco[13] categorized combustion models into three categories based on their uses in examining different engine problems. The categories are the zero-dimensional (or thermodynamic) model, the quasi-dimensional (or entrainment) model, and the multi-dimensional (or detailed) model.

3.2.1.1 Zero-dimensional Model[11]

The zero-dimensional model is structured around a thermodynamic analysis of the engine cylinder contents during the cycle. The assumptions include one-dimensional flow, isentropic adiabatic flow through nozzles simulating flow past valves, and unburned mixtures as mixtures of air, fuel vapor and residual gases. Specific heats of the gas mixture are modeled using polynomial functions of temperature. Compression is assumed to be adiabatic. Combustion assumes thermochemical equilibrium and progressive burning via mass elements. The expansion process assumes thermochemical equilibrium.

Heat transfer is modeled using correlations between the Nusselt, Prandtl and Reynolds numbers from heat transfer in steady turbulent flow over flat plates and pipes. These relationships are in the form of:

$$Nu = aRe^{b}Pr^{c}$$

where a, b, and c are obtained from experimental data for a specific engine.

The combustion process is generally modeled from an apparent heat release or an experimentally obtained fuel burning rate. One of the most widely used correlations is based on the Wiebe Function. In this function, the fuel burned is expressed as a fraction of the total fuel injected.[5]

$$FB = 1 - \exp[-K_2(t)^{(K_1+1)}]$$

where FB = fraction of fuel burned/total injected

t = time from ignition

K_1 = shape factor for combustion curve

K_2 = combustion efficiency coefficient.

Another typical function form is the cosine function:[11]

$$X(\theta) = (1/2)\{1 - \cos\pi[(\theta - \theta_o)/\Delta\theta_b]\}$$

where $X(\theta)$ = mass fraction burned at crank angle θ

θ_o = crank angle at the start of combustion

and $\Delta\theta_b$ = burn duration.

There are numerous other combustion models that utilize various heat release patterns. Some replace the heat release curve with two straight lines. In this type of combustion model, one line simulates the rapid combustion of the bulk of the injected fuel and the other line represents the slower combustion phase further down the expansion stroke.

An empirical model developed by Whitehouse and Way[14] is based on elementary combustion principles. Fuel is assumed to be prepared for combustion as a result of fuel-air mixing. The reaction rate calculates the burn rate in the premixed stage of combustion. The preparation rate becomes governing during the diffusion burning phase as the fuel is assumed to burn as rapidly as it is prepared. (The Whitehouse and Way model will be dealt with in detail in a later chapter.)

In general, thermodynamic combustion models are useful when performing a design trade off or comparison analysis to evaluate the effects of change in engine design and operation. Since, however, the details of the combustion process are an input to the model, the results can only indicate what will transpire if the engine burns in the specified manner. These models cannot address the feasibility of the engine operating in the prescribed manner because the details of the burning process are not linked to the engine design and operation.[15]

3.2.1.2 Quasi-dimensional Model[11]

Quasi-dimensional models are also structured around a thermodynamic analysis of the engine cylinder during the cycle. Many of the same assumptions are utilized to describe the various portions of the process as are used in the thermodynamic model. The combustion process, on the other hand, is based on more fundamental physical quantities such as turbulent intensity, turbulent mixing, jet characteristics in jet mixing and the kinetics of the fuel-oxidation process.

The quasi-dimensional models can be utilized for the same purposes as the zero-dimensional models except that they can now be used where changes in the combustion process can be a dominant factor. The major drawback of the quasi-dimensional model is its inability to examine, in detail, the interaction between fluid flow and engine geometry.{14}

3.2.1.3 Multi-dimensional Model{11}

In a multi-dimensional model, the governing partial-differential equations describing conservation of mass, momentum, energy and species, and the sub-models describing turbulence, chemical kinetics, and etc. are numerically solved subject to boundary conditions and other restraints. These models have the potential for examining the interaction between fluid flow and engine geometry that is lacking in the quasi-dimensional model. The detailed model will predict engine performance and emission characteristics from the first principles with virtually no empirical relationships. Unfortunately, solving the relevant conservation equations in three-dimensional, time dependent formulation, coupled with the state equations and sub-models leads to a computer program that will tax even the most capable computer system.

Chapter 4

THE TWO CYLINDER CYCLE COMBUSTION MODEL

Since the two cylinder cycle reciprocating engine is a totally new concept, combustion modeling can be even more difficult than for a compression ignition engine. However, the approach taken models the expansion cylinder of the new cycle after a diesel engine cylinder. The beginning of the expansion stroke will simulate a diesel engine with its piston at TDC with a charge of air. For this initial combustion model, the air will be assumed to be contained within the cylinder, at pressure, with no additional air added after expansion, as in the actual new engine cycle.

4.1 Assumptions

The assumptions for this single zone combustion model are essentially those previously mentioned for the thermodynamic type of models.

a. The First Law of Thermodynamics is used to establish an energy balance to determine the temperature at the end of each step.

b. The working fluid is treated as an ideal gas.

c. The system contents are homogeneous and of uniform temperature and pressure.

d. The changes in gas properties due to the rate of change of the gas composition are considered to be negligible.

e. Combustion is treated as a reversible heat release process.

f. Combustion products are formed in the proportions according to the law of perfect combustion.

g. No dissociation of the products of combustion occurs.

h. Only four gases are considered to be present and are varied as required for perfect combustion.

i. The incoming air charge is assumed to be pure air plus a fraction of the residual gases remaining in the cylinder.

4.2 Thermodynamics of Internal Combustion Engines

4.2.1 Ideal Gas[16]

The assumed thermally ideal gas obeys the state equation

$$pV = M\bar{R}T$$

where p = pressure

V = volume

M = number of moles

$\bar{R}$ = universal gas constant

and T = temperature.

The specific gas constant, R, can be written in terms of $\bar{R}$ and m_w, the molecular weight of the gas.

$$R = \bar{R}/m_w.$$

If the mass of the gas, $m = Mm_w$, then the state equation can be written as:

$$pV = mRT.$$

The specific internal energy for an ideal gas can be represented as a function of temperature:

$$u = f(T)$$

where u = specific internal energy

and f(T) = function of temperature dependent on the gas.

If the function f(T) is expressed in the form of a limited power series, then{17}

$$u = u_o + \sum_{n=1}^{n=5} a_n T^n$$

where a_1 to a_5 are constants which vary depending on the gas

and u_o = internal energy at absolute zero.

The specific heat at constant volume can be defined as:

$$C_v = (dq/dT)_v = (du/dT)_v.$$

Thus, following the same procedures as for the internal energy, above:{17,18}

$$C_v = \sum_{n=1}^{n=5} n a_n T^{n-1}$$

The specific enthalpy, h, for an ideal gas is given by:

$$h = u + RT.$$

It follows that:{17,18}

$$h = h(T) = u_o + \sum_{n=1}^{n=5} a_n T^n + \bar{R}T.$$

At absolute zero, T=0:

$$h = h_o = u_o.$$

Therefore, for a perfect gas, the internal energy varies linearly with temperature as:

$$h = h_o + C_v T + \bar{R}T.$$

The specific heat at constant pressure, C_p, is defined by:

$$C_p = (dq/dT)_p = (dh/dT)_p.$$

For a perfect gas:

$$C_p = C_v + \bar{R}.$$

Now, enthalpy can be expressed by:

$$h = h_o + C_p T.$$

For thermodynamic processes with gases of constant composition and specific heats undergoing state changes;

$$h_o = u_o = 0.$$

Then,

$$u = C_v T;$$

$$h = C_p T;$$

$$h - u = (C_p - C_v)T = \bar{R}T;$$

and $C_p - C_v = \bar{R}$.

Gas data are often given in terms of enthalpy vice internal energy. The conventional form is:

$$h(T)/RT = (h - h_o)/RT$$
$$= a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4.$$

and the internal energy is expressed as:

$$u(T)/RT = (a_1-1) + a_2T + a_3T^2 + a_4T^3 + a_5T^4.$$

The values for the polynomial coefficients, a_0 to a_5 are provided in Table 2. Other formulations for the calculation of enthalpy and specific heat are available in the literature.{5,26,27}

Table 2: Polynomial Coefficients

Range: 500 - 3000 Degrees Kelvin

	a_1	a_2	a_3	a_4	a_5
CO_2	3.0959	2.73114E-03	-7.88542E-07	8.66002E-11	0.0
H_2O	3.74292	5.65590E-04	4.95240E-08	-1.81802E-11	0.0
O_2	3.25304	6.52350E-04	-1.49524E-07	1.53897E-11	0.0
N_2	3.34435	2.94260E-04	1.95300E-09	-6.57470E-12	0.0
C_8H_{18}	-0.71993	4.6426E-02	-1.68385E-05	-2.67009E-09	0.0
C_3H_8	1.13711	1.45532E-02	-2.95876E-06	0.0	0.0

4.2.2 Properties of Gas Mixtures{18}

Mixtures of gases obey the following.

a. The gas mixture as a whole obeys the equation of state, $pV = MRT$, where M is the total number of moles of all species.

b. The total pressure of the mixture is equal to the sum of the pressures which the individual components/species exert.

c. The internal energy, enthalpy and entropy of the mixture equals the sum of the internal energies, enthalpies and entropies which each individual component/species would have if it separately occupied the

entire volume of the mixture at the same temperature.

Thus, for mixtures of ideal gases the mole fraction is given by:

$$x_i = M_i/M$$

where M_i = moles of a specie

and M = total number of moles.

Then,

$$\sum x_i = 1.0.$$

Enthalpy is given by:

$$H = \sum M_i h_i = M \sum x_i h_i.$$

Internal energy is given by:

$$U = \sum M_i u_i = M \sum x_i u_i.$$

Specific Heats are given by:

$$C_p = \Sigma\, x_i C_{pi}$$

$$C_v = \Sigma\, x_i C_{vi}.$$

4.2.3 The First Law of Thermodynamics{17}

The emphasis of this model is the closed portion of the cycle. Therefore, the First Law of Thermodynamic for closed systems is simply:

$$dQ - đW = dU'$$

where dQ = heat energy transfer

$đW$ = work energy transfer

dU' = change in internal energy.

The internal energy is defined by:

$$U' = U + KE + PE$$

where U = the intrinsic internal energy

KE = kinetic energy

PE = potential energy.

For a closed system, we can assume that PE = KE = 0. Therefore,

$$dQ - dW = dU$$

where $U = M \Sigma x_i u_i$

M = total number of moles

x_i = mole fraction of gas i

u_i = specific internal energy of gas i.

For non-reacting closed systems, we can write:

$$dQ - dW = dU$$

where $dW = pdV = (\Sigma x_i p) dV$

and $dU = Md(\Sigma x_i u_i)$.

For a reacting closed system, we can expand this to:

$$dQ - pdV = (U_{op} - U_{or}) + U_p(T) - U_r(T)$$

where $(U_{op} - U_{or}) = \Delta U_o$

ΔU_o = heat of reaction

$U_p(T)$ = energy of products as a function of time

$U_r(T)$ = energy of reactants as a function of temperature.

4.3 Heat Transfer from the gas to the Cylinder

To be able to balance the energy in a real system, the heat transfer from the combustion gas to the walls of the cylinder must be considered. Two basic equations are generally accepted for use in cycle

calculations. These are the correlations developed by Annand and Woschni. The relationship by Woschni[19] is based upon a forced convection model.

$$q/A=C_3 d^{-0.2} p^{0.8} T_g^{-0.053} (C_1 V_p+(C_2 (p-p_o) VT'/p'V')^{0.8} (T_g-T_w)$$

where C_1, C_2, and C_3 = constants

A = area

D = cylinder bore

p = pressure

T_g = mean gas temperature

T_w = wall temperature

V_p = piston velocity

p_o = motoring pressure

p' = trapped pressure

V' = trapped volume

T' = trapped temperature.

Although Woschni's expression is readily accepted, it does not separately distinguish between convection and radiation.

The Annand equation is also largely based on turbulent convection. Unlike the Woschni correlation, Annand claims that the Reynolds number is the major parameter affecting convection. Convection is the first term in his equation. The second term in Annand's equation is a radiation term assuming grey body radiation. Thus:[20]

$$q/A = a(k/D)(Re)^b (T_g-T_w) + c(Tg^4-Tw^4)$$

where q = heat transfer rate

A = area

a, b, c = constants

k = thermal conductivity

D = bore

Re = Reynolds Number = $\rho V_p D/\mu$

ρ = density

V_p = piston velocity

μ = viscosity

Tg = temperature of gas (mean)

Tw = temperature of wall.

The range of values for Annand's constants are:

for a four stroke engine:

$a = 0.26$

$b = 0.75 \pm 0.15$

$c = 3.88 \pm 1.39 \times 10^{-8}$ J/sm^2K^4

for a two stroke engine:

$a = 0.26$

$b = 0.64 \pm 0.10$

$c = 3.03 \pm 1.06 \times 10^{-8}$

Since Annand's equation separates the convective term from the radiation term, it is believed that the Annand correlation is better suited to the new cycle calculations.

4.4 The Combustion Model

In the process of heat release from combustion, both physical and chemical effects are involved. Liquid fuel injected into an engine must be heated, vaporized, and mixed with oxygen in the preparation process prior to combustion. Once the fuel is prepared, it may then burn at a rate controlled by chemical kinetics. It has been demonstrated that the time required for combustion of the prepared fuel is negligible as compared to the preparation time.

At the beginning of the burning period, chemical kinetics are important due to the low temperatures. When fuel is first injected into a cylinder of a diesel engine, the temperature is generally such that rapid burning will not occur. Additionally, the heat transferred to the incoming fuel causes the temperature to drop in the cylinder. As the temperature rises in the cylinder, the combustion rate rises, thus increasing the temperature. The heat release rate continues to rise until the lack of prepared fuel becomes the controlling factor. When the excess prepared fuel is depleted, combustion will proceed at the rate of fuel preparation. Figure 12 represents the effects of preparation rate and reaction rate in premixed burning as a function of crank angle.

4.4.1 Preparation of Fuel

After injection, the fuel is physically prepared for combustion. As mentioned before, this process involves the atomization, vaporization and mixing of the fuel with air. The rate of preparation can be assumed to be proportional to the total surface area of the fuel spray droplets. If all the droplets are assumed to be of identical size, then it

follows:[7,14,21]

$$M_i = np\pi D_o^2/6$$

$$M_u = np\pi D^2$$

where M_i = Mass of fuel injected

M_u = Mass of fuel unburned

n = number of fuel droplets

p = fuel droplet density

D_o = Initial droplet diameter

D = Droplet diameter.

The total area

$$\text{Area} = n\pi D^2 = n\pi(6M_u/np\pi)^{2/3}$$

$$\text{Area} = (6M_i/npD_o^3)^{1/3}(6M_u/p\pi)^{2/3}$$

$$\text{Area} = 6M_i^{1/3}M_u^{2/3}/pD_o.$$

Assuming that the density, p, and initial diameter, D_o, are constant, then the

$$\text{Area} \;\alpha\; M_i^{1/3}M_u^{2/3}.$$

Allowing for the effect of oxygen availability on the mixing of the fuel, the preparation rate, PR, can be written as:

$$PR = KM_i^{1-x}M_u^{x}P_{O2}^{m}$$

where x = empirical constant

m = empirical constant

P_{O2} = partial pressure of oxygen

K = constant.

The constant K is a function of the characteristics of fuel injection, air movement and combustion chamber shape. Typical values for four

stroke engine are:[14]

$$K = 0.008 - 0.020$$

$$x = 2/3$$

$$m = 0.4.$$

4.4.2 Reaction of Fuel

Since diesel fuel is not a pure substance, it is impossible to ascertain the exact chemical equations involved since the actual compounds in the fuel are unknown. The temperatures that are available from experiments are only average cylinder temperatures. With these approximations/estimations, the equations for reaction rate are highly empirical. The degree of approximation involved may be justified due to the short time period during which chemical kinetics is of importance. Also, the total fuel that is burned is equal to the amount of fuel that is prepared. The reaction rate equation that was proposed by Whitehouse and Way [7,14,21] is based on the Arrhenius equation.

$$\dot{R} = (K'P_{O2})/(N\sqrt{T}) \int (PR-R)\,dx \; \exp(-act/T)$$

where R = reaction rate per degree crank angle

K' = empirical constant

act = empirical constant

P_{O2} = partial pressure of oxygen

PR = preparation rate

N = engine speed in rpm

T = cylinder temperature.

The effect of the ignition delay period is incorporated in the Arrhenius type expression $\exp(-act/T)$. Typical values of K' and act are :

$act = 1.4 \times 10^4$

$K` = 1.2 \times 10^{10}$ for two stroke engines

$K` = 65 \times 10^{10}$ for four stroke engines

4.5 Verification of the Model

The model was converted top computer code using TRS-80 Model III Disk Basic. The program listing is presented in Appendix B.

In an effort to set the empirical coefficients, the average value was used for all coefficients that had a range of values for four stroke engines. The program was run and compared to the data obtained by Remley{22} in actual engine testing in the Sloan Automotive Laboratory. Figure 13 represents the pressure versus volume curve for the model and for the engine run by Remley. Appendix A provides specifications of the test engine.

Chapter 5

SELECTION OF FUEL INJECTION POINT

In order to obtain the maximum work and highest efficiency from the new two cylinder cycle, the time of fuel injection should be optimized. To obtain this optimum, a number of cycles were run on the computer.

5.1 Selection of the Model Coefficients

The model was run assuming the expansion cylinder at TDC with an air charge at a temperature and pressure of $1090^{o}K$ and 10 atmospheres while the engine speed of 850 rpm and air/fuel ratio were held constant. The selected fuel was C_8H_{18} (iso-octane) with a lower heating value of 4.2×10^{7} joules/kilogram and a residual air fraction of 0.05.

The model was run several times to obtain a value of K in the equation:

$$PR = KM_i^{(1-x)} M_u^{x} P_{O2}.$$

The values of x and m were held constant at 2/3 and 0.04, respectively, as the values used for four-stroke diesel engines. When searching for a value of K, a diffusion combustion period of 70 - 120 degrees of crank angle was sought. This was found through several iterations to occur at a value of K = 0.012.

The values of constants for the reaction rate equation:

$$R = [K'P_{O2}/NT^{0.5}]\exp(-act/T)\int(PR-R)dx,$$

were selected as the values for four-stroke diesel engines.

With this input data and selection of constants, the model yields a heat release rate curve which closely resembles that described by Ricardo, Lyn and Whitehouse et al, see figure 14. The premixed burning phase yields approximately 45 percent of the heat release, the diffusion controlled burning phase yields approximately 45 percent of the heat release with the tail of combustion providing the remaining 10 percent.

5.2 <u>Optimizing Fuel Injection</u>

Intuitively, the maximum work and highest efficiency would be expected with fuel injection and combustion occuring at TDC, or immediately thereafter. This, however, does not appear to be the case when the data is evaluated. See figures 15 through 21. While the fuel injection is varied from 180 (TDC) to 205 degrees crank angle, with an injection period of 20 degrees, the thermal efficiency rises. For fuel injection occurring from 180 to 195 degrees, the temperature at 360 degrees (BDC) is not sufficient to heat the regenerator matrix to a temperature which will pre-heat the incoming air charge to 1090 degrees Kelvin as specified by the input data. For fuel injection occuring at 205 degrees, and later, incomplete combustion will result.

From this approach, the optimum point of fuel injection occurs at 200 degrees crank angle for a fuel injection period of 20 degrees.

When the fuel injection period is reduced to 10 degrees, a similar pattern is observed. A fuel injection point on, or before, 200 degrees results in the cylinder gas temperature dropping too low to support sufficient air charge pre-heat. Fuel injection on, or after, 210 degrees results in incomplete combustion. See figures 22 through 24. In this case, the optimum point of fuel injection occurs at 205 degrees. The thermal efficiency for this case is higher that the case of a 20 degree injection period. Also, the specific fuel consumption is lower in the case of 10 degree injection as compared to 20 degree injection.

Through a similar analysis, the case of an air/fuel ratio of 25 yields an optimum fuel injection point of 195 degrees crank angle for a period of 20 degrees. For this air/fuel ratio, the value of K in the preparation rate equation was selected as 0.018 to achieve a similar heat release rate pattern.

Chapter 6

COMMENTS AND RECOMMENDATIONS

As can be readily seen, the output from this type of thermodynamic model is dependent on the value of the empirical coefficients. The characteristics of the heat release rate curve will shift as a function of air/fuel ratio, temperature, pressure and engine speed. Therefore, only a comparison of results from a defined heat release should be used for qualitative comparison analysis.

Since this computer model was written for a personal computer, the time required for one run is excessively long for a detailed comparison analysis. The run time for one run with five degree increments is approximately 1 hour and 20 minutes. A motoring analysis (no combustion) requires approximately 15 minutes. The amount of time required in the combustion iteration process is the difference between the two. These times were obtained when running the program with no remark statements and elimination of all unnecessary spaces in the program. Undoubtedly, the efficiency of the program can be somewhat increased by utilizing some clever programming techniques. However, the use of a small computer strictly dedicated to a comparison analysis with crank angle increments of one or two degrees can occupy the machine for an inordinate period of time.

6.1 Recommendations

This single zone model allows for cycle studies. However, a problem that must be explored is the formation of soot and gaseous pollutants. This can be accomplished by expanding the model to a two or four zone model.{23,24} During this model expansion, the effect of chemical kinetics should be further examined to display a more realistic combustion process. The values of the coefficients for the polynomial expression of enthalpy for the other products of combustion are readily available.{17,18}

The effects of heat transfer from the system may be more appropriately modeled by the use of the widely accepted Woschni correlations.{19} The use of Annand's correlation, however, does allow for the separation of convection and radiation.

The effects of mixing of the air charge with the fuel must be further explored to determine the effects on combustion intensity and efficiency.{25}

The use of a larger computer system would be most beneficial in a comparison analysis. Single runs can be easily done on a personal computer system, however, many runs using small crank angle increments are best, although more costly, performed on a main frame system capable of performing numerous simultaneous calculations.

Lastly, to obtain realistic coefficients for the empirical constants in the preparation and reaction rate equations, experiments using a rapid compression machine are considered appropriate. This

would provide for realistic data with minimum cost.

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Figures

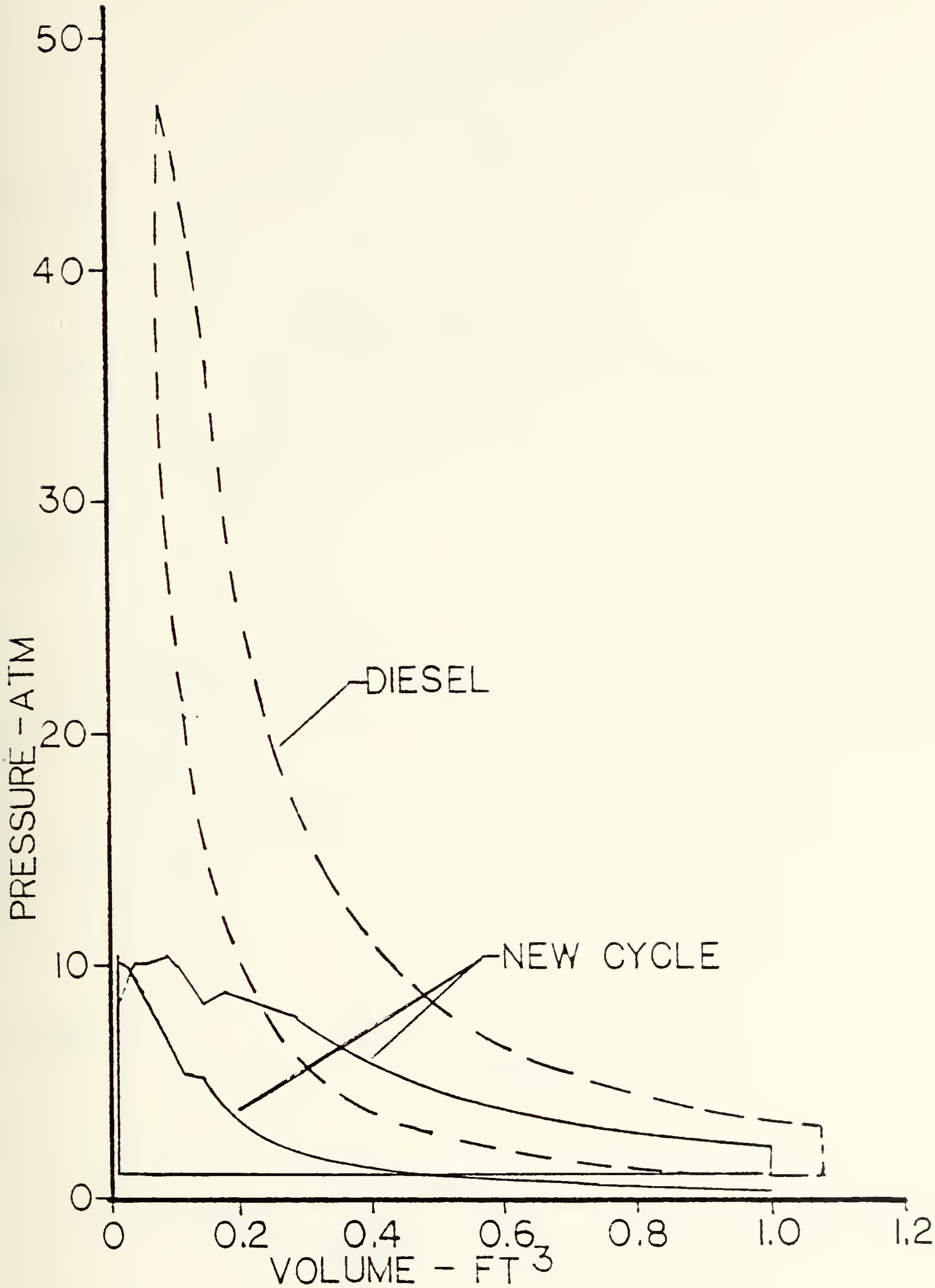


Figure 1: Pressure vs. Volume for New Cycle vs. Diesel Engine

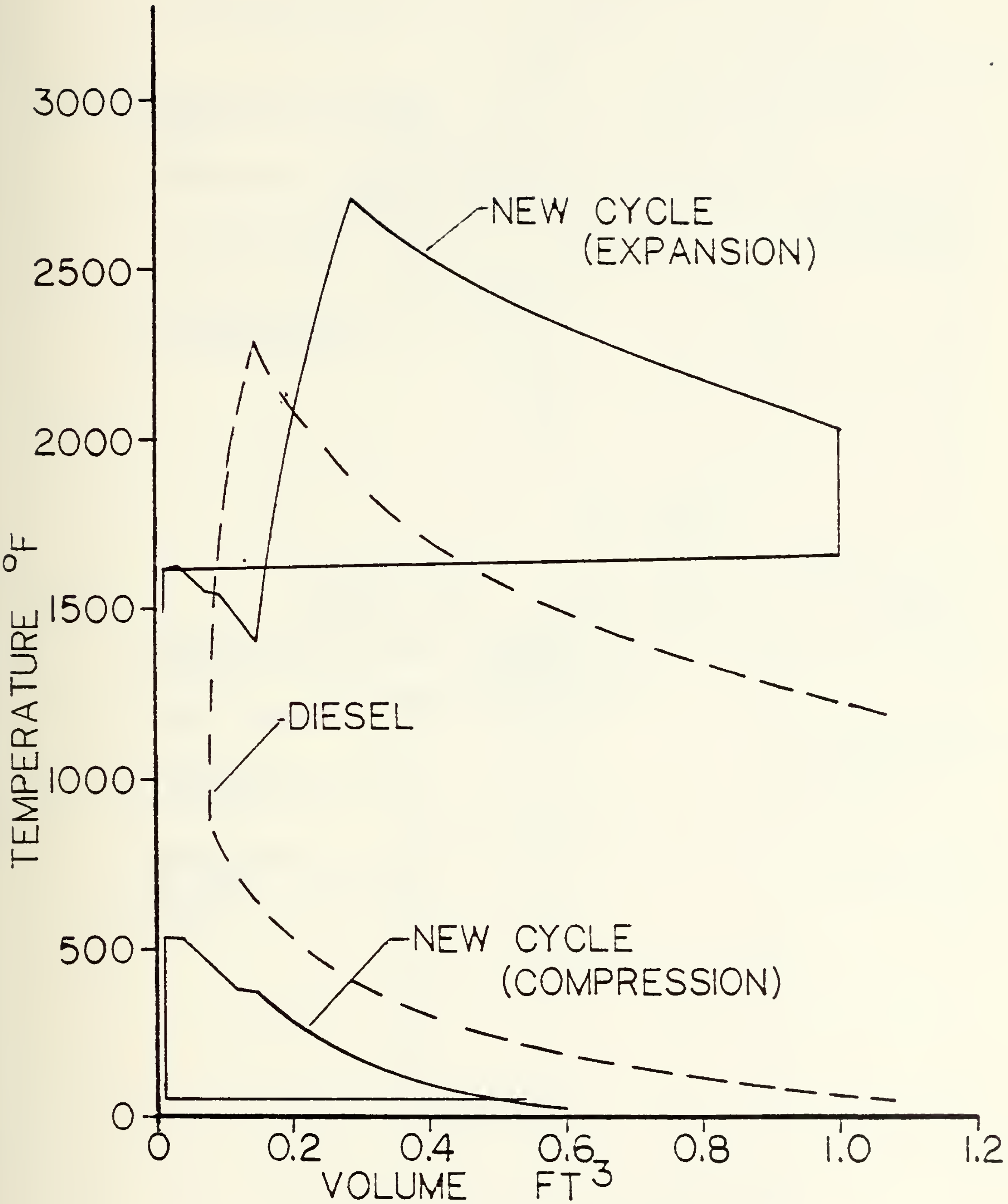


Figure 2: Temperature vs. Volume for New Cycle vs. Diesel

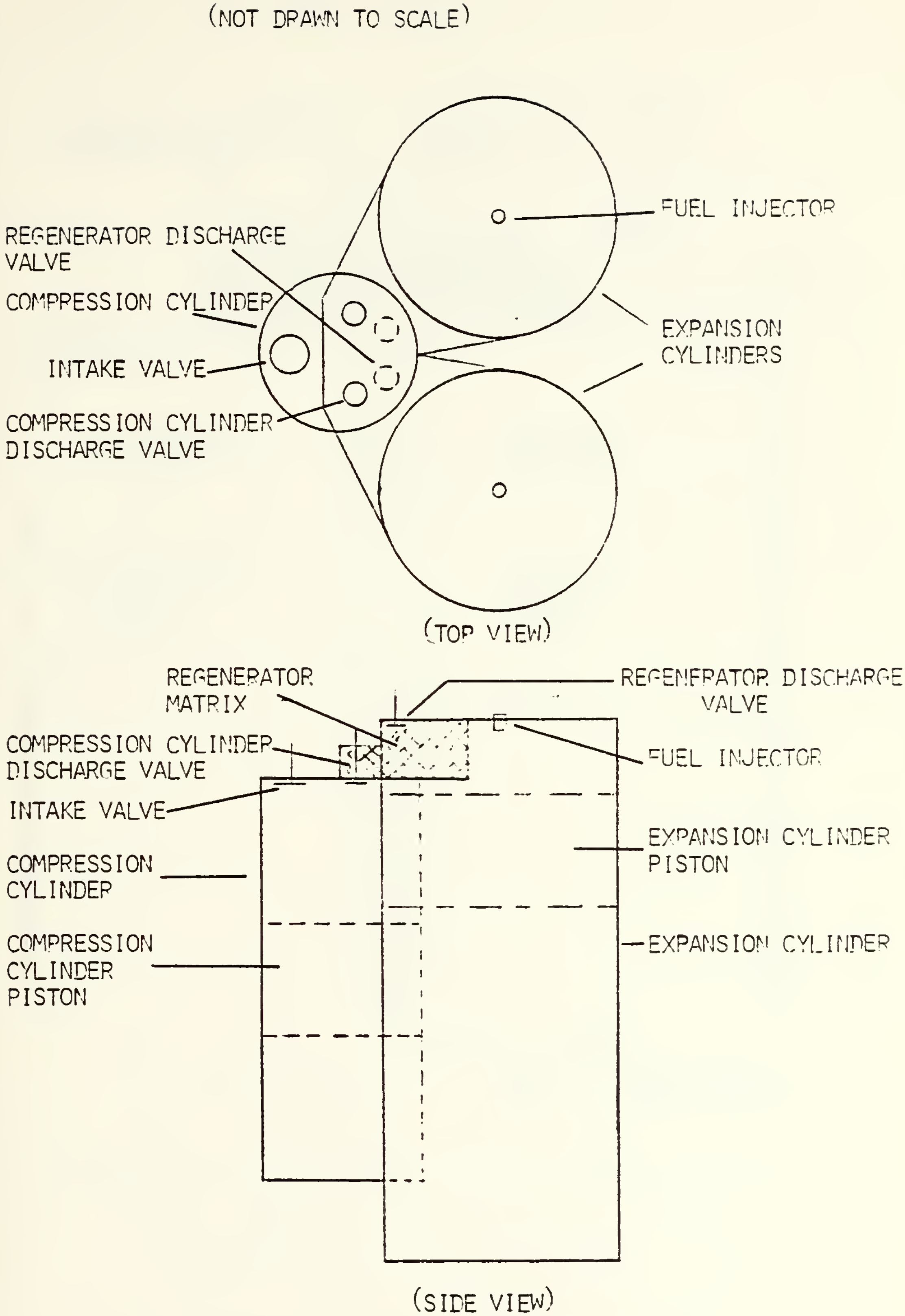
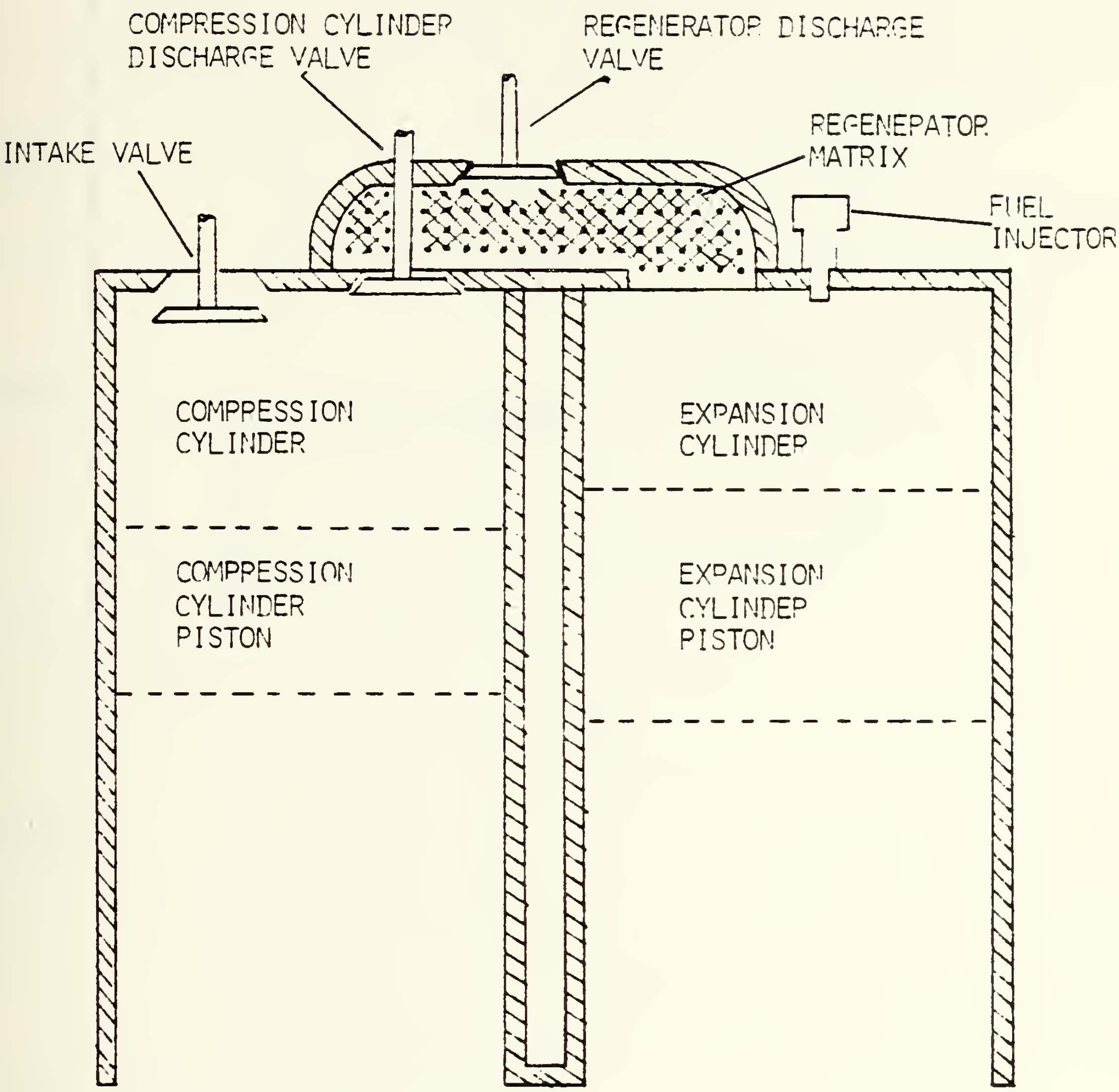


Figure 3: Potential Arrangement of Components for New Engine[2]

(NOT DRAWN TO SCALE)



Expansion Cylinder
Bore: 0.3725 meters
Stroke: 0.3725 meters

Figure 4: Cutaway View of New Cycle Engine[2]

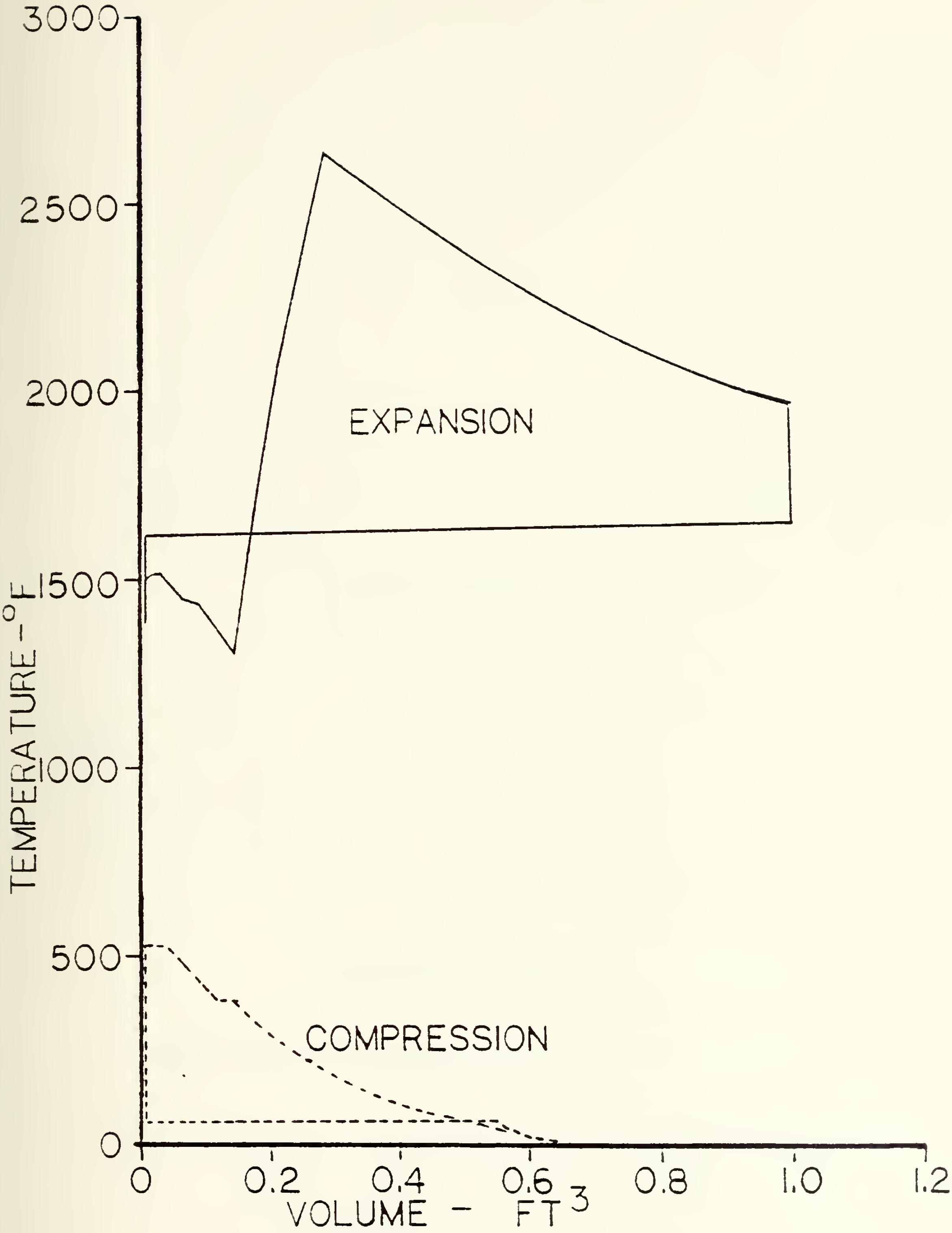


Figure 5: Temperature vs. Volume for New Cycle

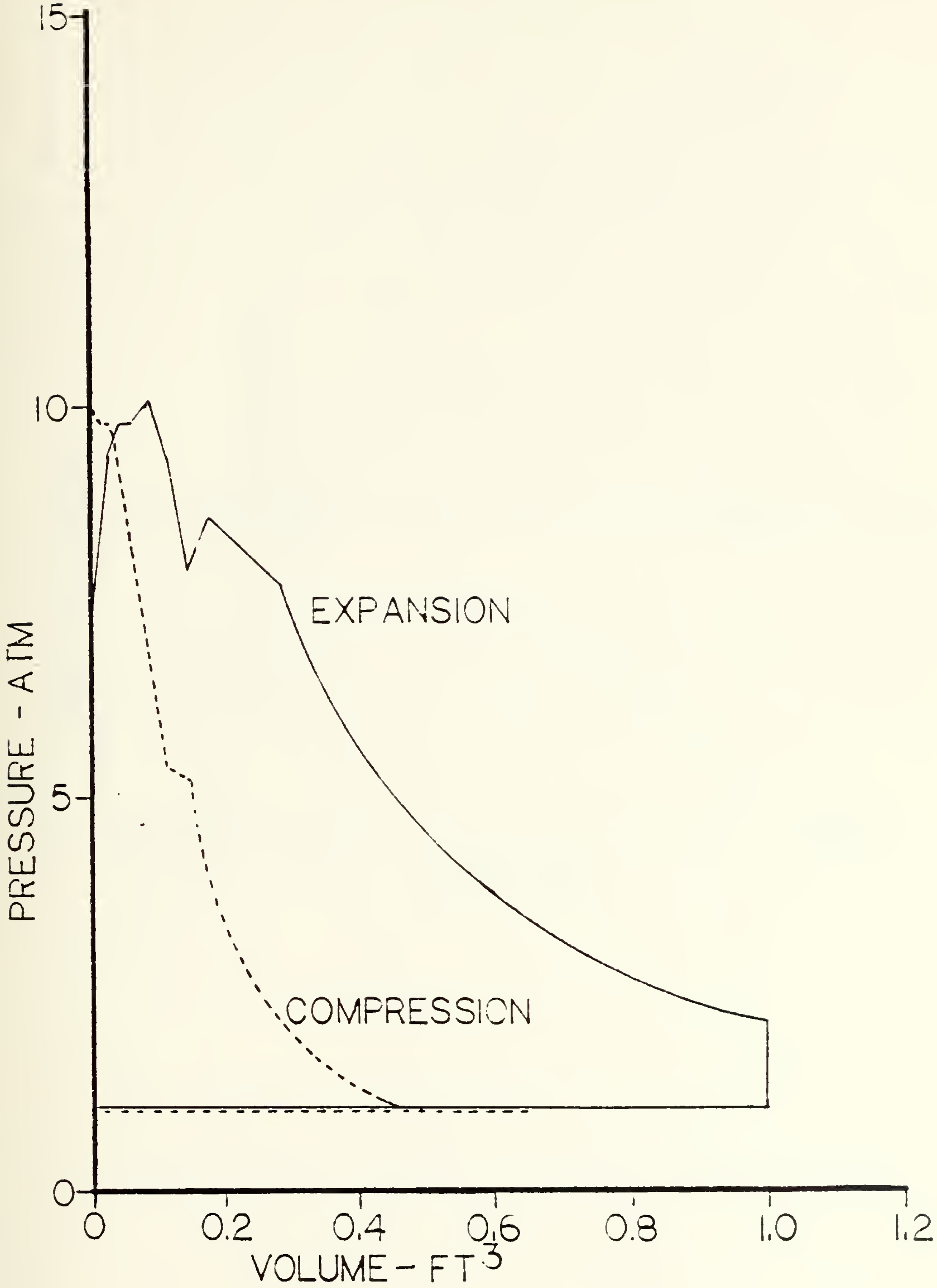


Figure 6: Pressure vs. Volume for New Cycle

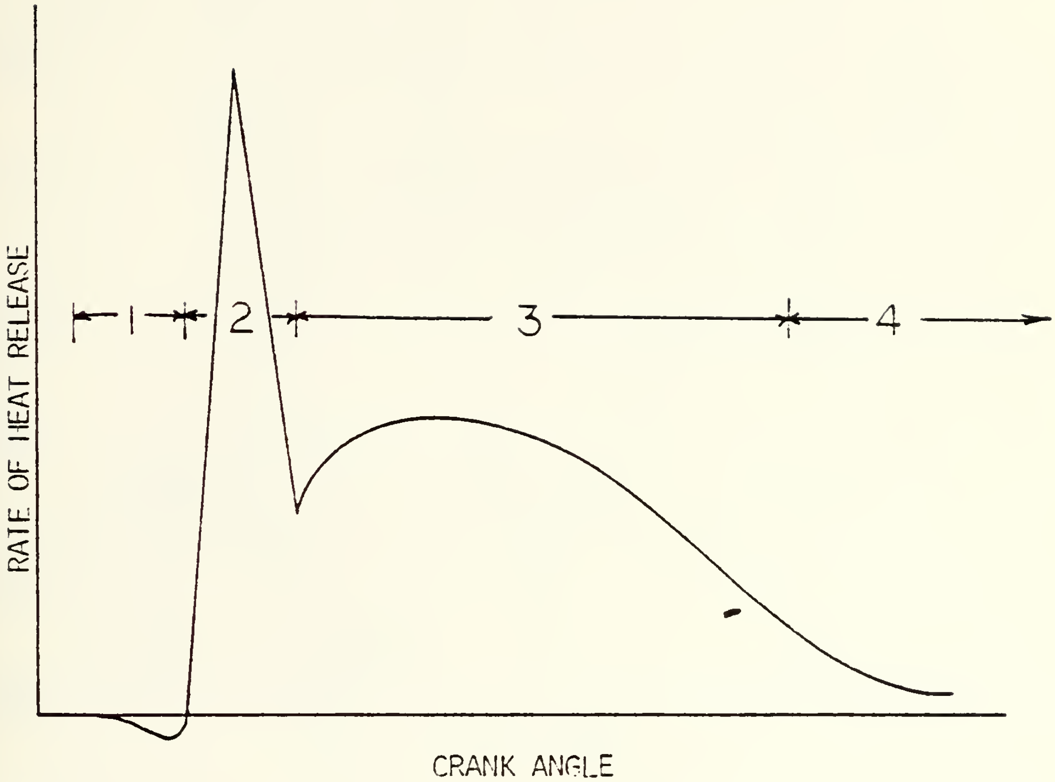


Figure 7: The Four Phases of Combustion

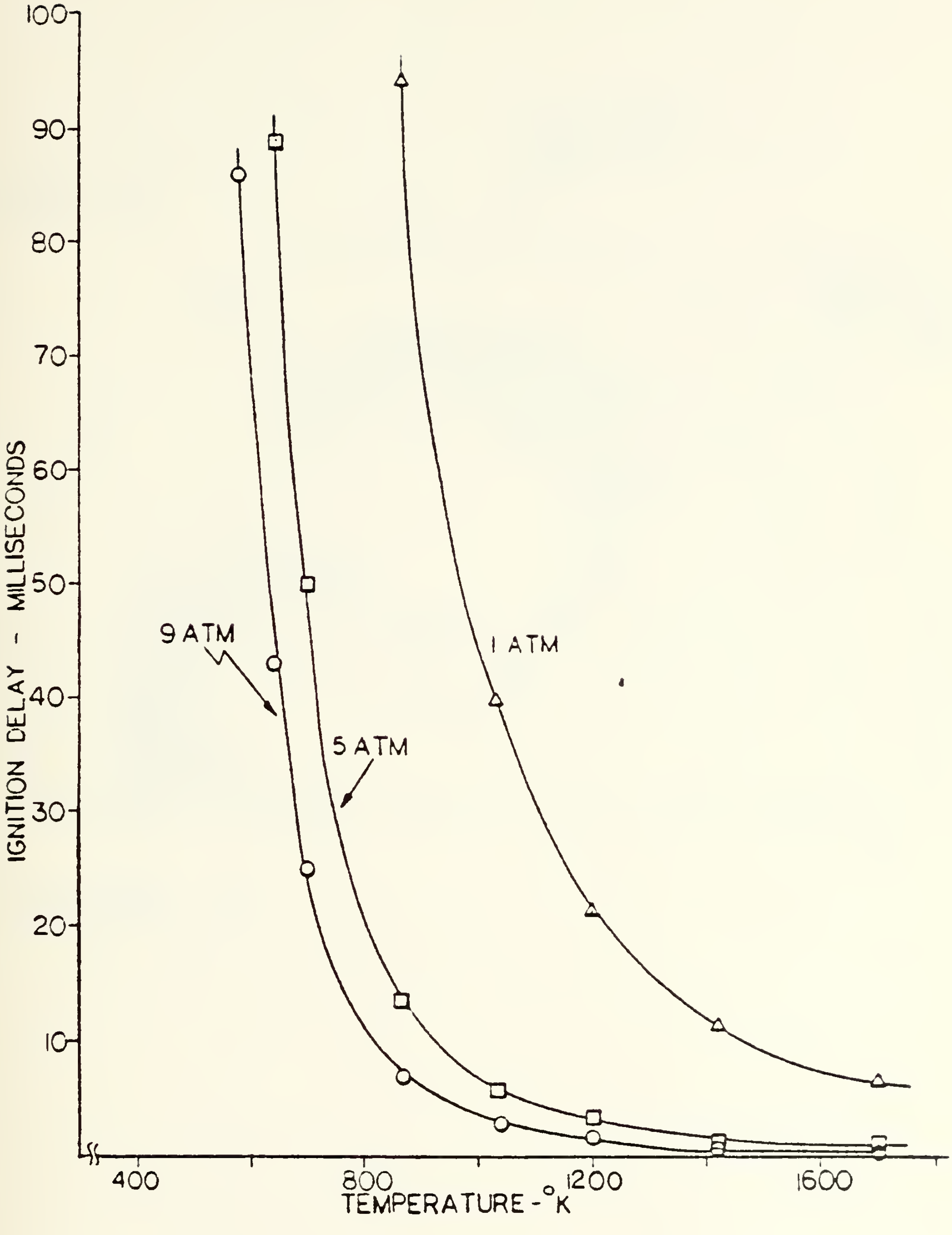


Figure 8: Effects of Temperature and Pressure on Ignition Delay

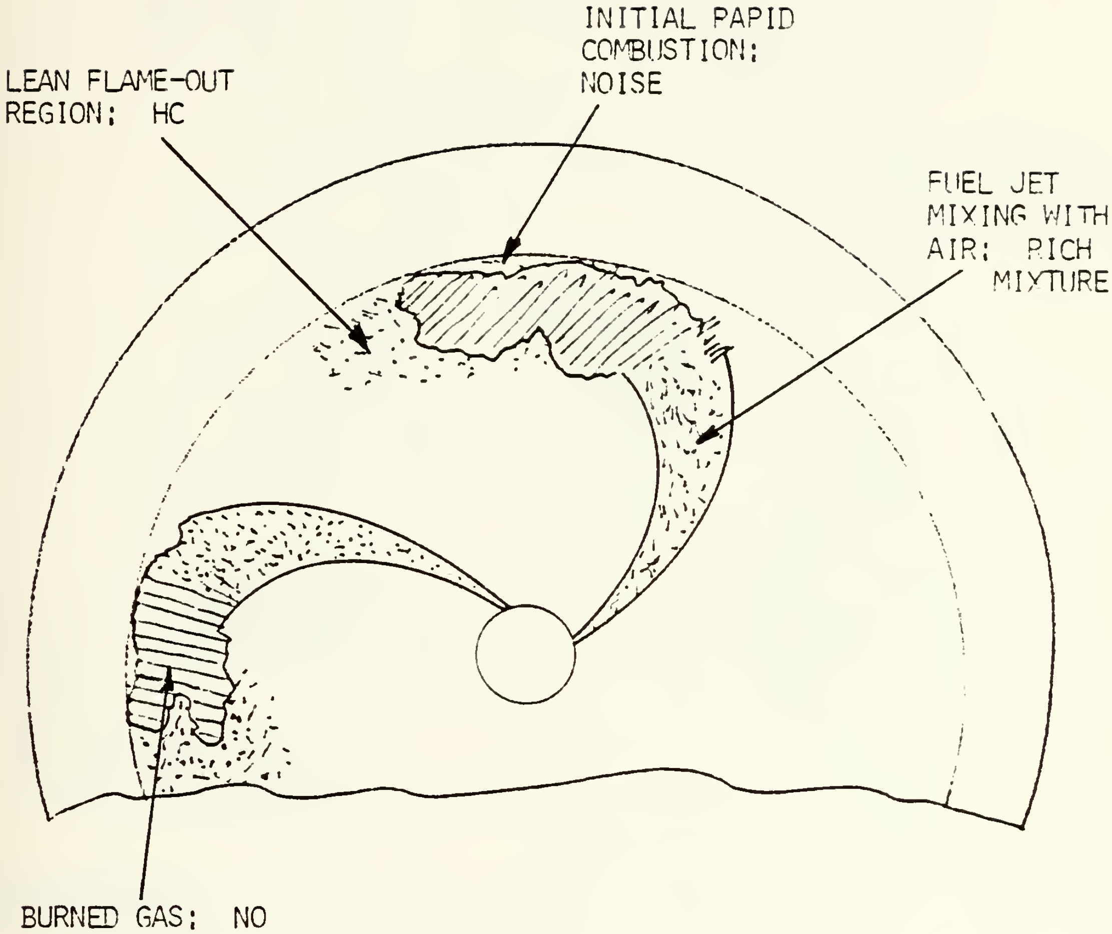


Figure 9: Direct-Injection Compression Ignition Engine [11]
Combustion during the PREMIXED Phase

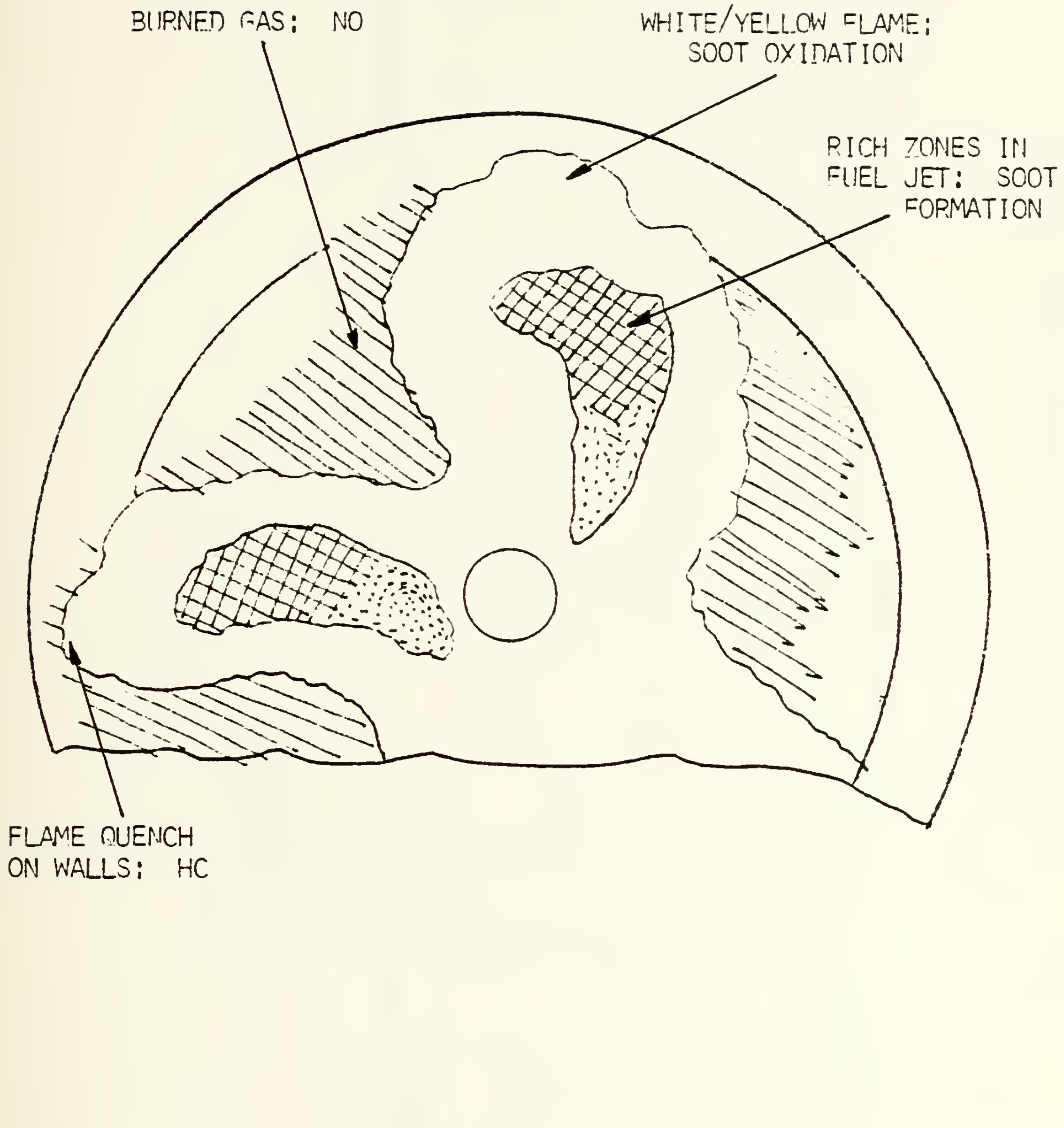


Figure 10: Direct-Injection Compression Ignition Engine Combustion during the MIXING CONTROLLED Phase (11)

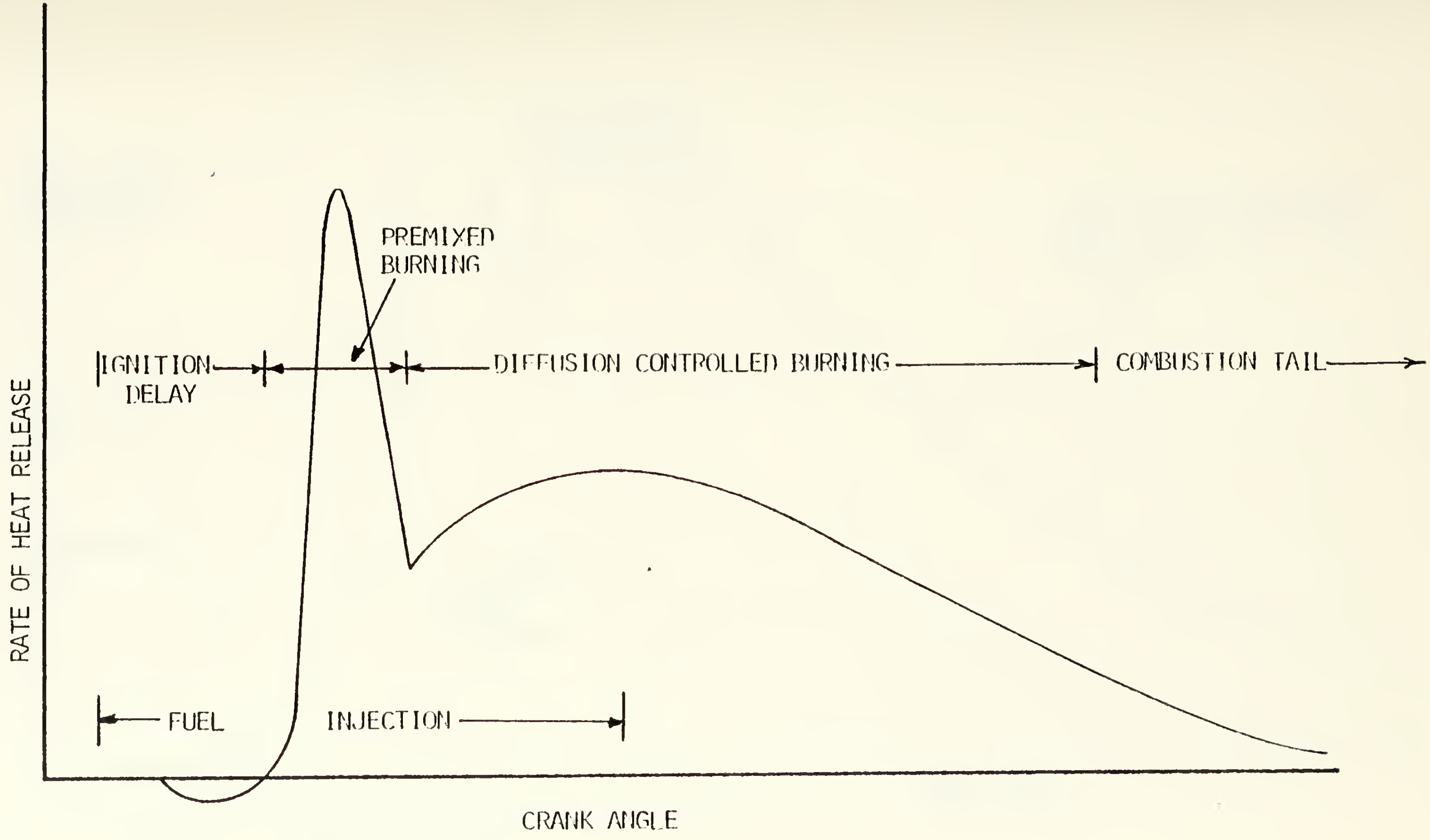


Figure 11: Typical Heat Release Diagram Showing Four Stages of Combustion

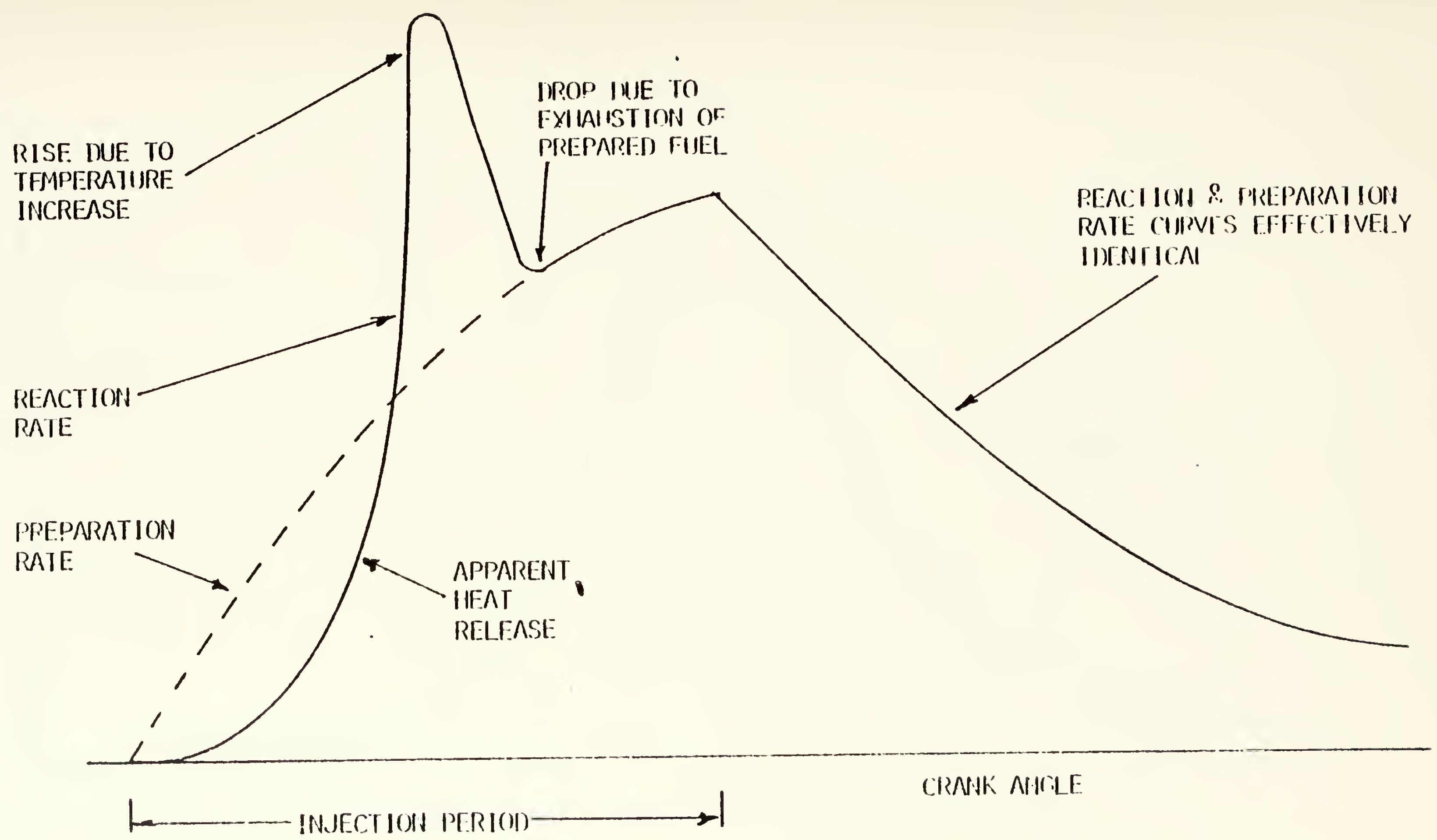


Figure 12: Heat Release Rates Calculated by Whitehouse Way Model

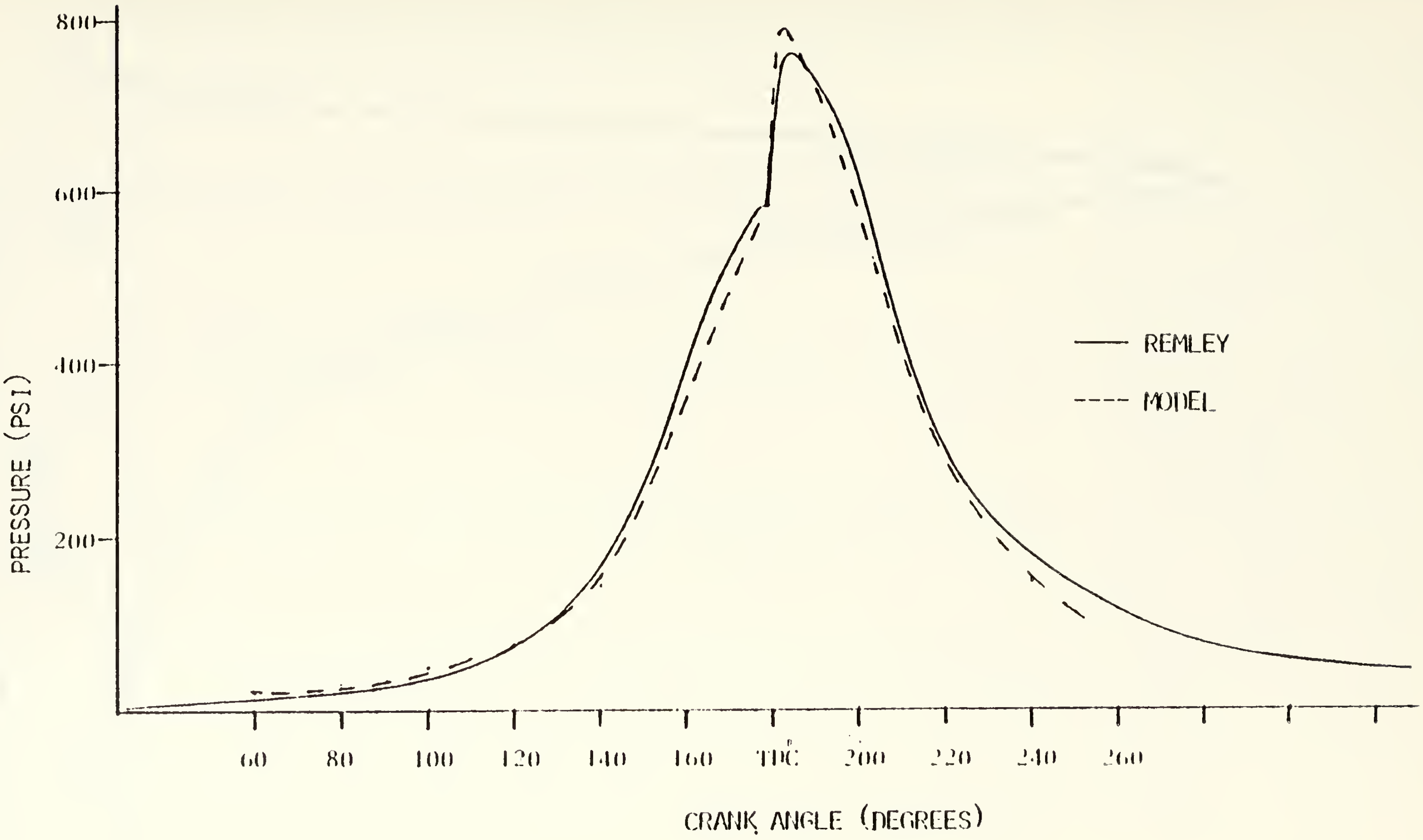


Figure 13: Comparison of Model with results from Remley Test[illegible]

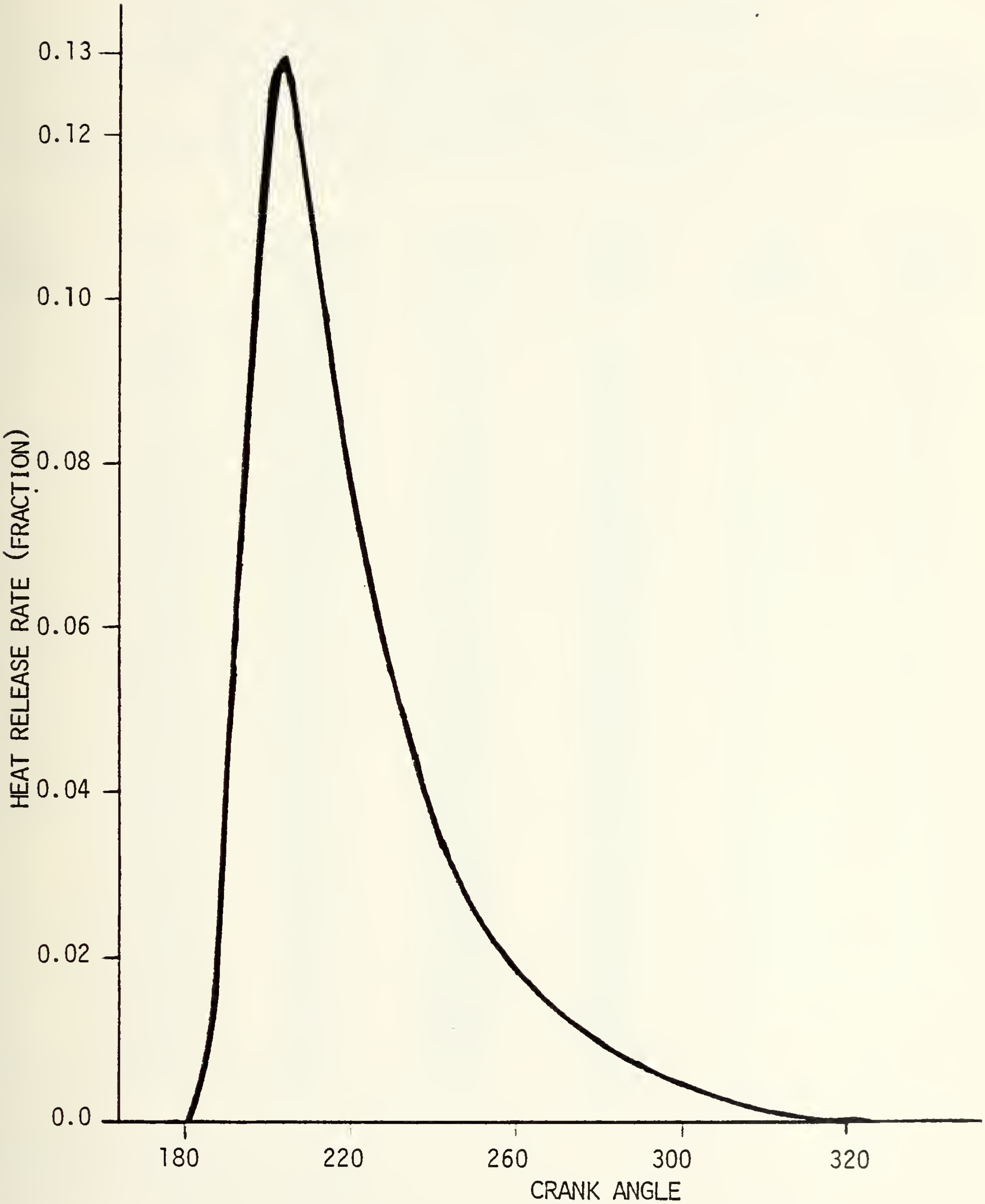


Figure 14: Heat Release Rate Obtained from Computer

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE - MOTORING

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .502836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M^3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.2021437	10.1335	1090	0	0	0
185	.2022035	10.0313	1084.35	55.4157	0	0
190	.2023525	9.75535	1074.87	222.45	0	0
195	.2025721	[illegible]	1061.64	507.301	0	0
200	.2028754	[illegible]	1044.51	875.311	0	0
205	.203255	8.30032	1025.23	1331.83	0	0
210	.2038011	7.74307	1002.63	1837.64	0	0
215	[illegible]	7.13275	[illegible]	2394.75	0	0
220	.2056433	6.52443	957.311	2981.21	0	0
225	[illegible]	5.94235	922.353	3569.51	0	0
230	.2088864	5.38135	905.483	4202.23	0	0
235	[illegible]	4.83258	884.135	4811.33	0	0
240	.2160718	4.41811	862.27	5411.27	0	0
245	.2187555	3.99245	837.341	5991.87	0	0
250	.2212317	3.62454	815.717	6550.14	0	0
255	.2227393	3.2311	795.33	7084.33	0	0
260	.2244233	2.93516	775.358	7595.21	0	0
265	.2261133	2.7335	755.783	8081.94	0	0
270	.2278273	2.50413	735.713	8553.01	0	0
275	.2295558	2.32192	722.531	8981.39	0	0
280	.2314715	2.124	708.555	9383.48	0	0
285	.2332955	1.95773	694.439	9758.85	0	0
290	.2351155	1.8227	681.432	10123.4	0	0
295	.2369129	1.71073	669.375	10432.5	0	0
300	.238869	1.60507	658.305	10711.6	0	0
305	.2400674	1.51433	648.107	10970.8	0	0
310	.2418901	1.42591	639.254	11215.3	0	0
315	.2435197	1.35755	630.838	11432.7	0	0
320	.2449095	1.30585	623.572	11632.7	0	0
325	.2463036	1.25933	617.159	11795.5	0	0
330	.2473672	1.21778	611.832	11933.7	0	0
335	.2483372	1.1833	607.207	12043.8	0	0
340	[illegible]	1.15531	603.321	12147.5	0	0
345	[illegible]	1.1355	600.383	12213.1	0	0
350	.2503554	1.12036	598.306	12270.3	0	0
355	.2505438	1.11316	597.251	12305.1	0	0
360	.2507433	1.10937	596.547	[illegible]	0	0

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = [illegible] BARS
POWER (4 STROKE) = [illegible] KILOWATTS
SPECIFIC FUEL CONSUMPTION = 0 KG/KW-HR
THERMAL EFFICIENCY = [illegible] PERCENT

Figure 15

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 350 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
	FUEL INJECTION START AT 180		COMBUSTION COMMENCED			
180	.0101467	12.1025	1090	0	0	0
185	.0103066	12.3717	1121.73	53.4853	.0374117	.0374117
190	.0105805	12.5152	1193.67	244.477	.071894	.109305
195	.0105702	11.5312	1327.42	570.337	.102832	.212139
200	.0110754	12.1007	1426.41	1052.35	.129154	.341553
	FUEL INJECTION STOP AT 200					
205	.011555	12.2167	1435.89	1557.83	.112392	.453544
210	.0122312	11.2042	1522.14	2450.81	.0930122	.546357
215	.0129824	11.1335	1523.54	3312.01	.0755979	.624343
220	.0136432	10.0312	1501.76	4134.35	.0655128	.689455
225	.014615	9.434	1475.73	5193.74	.0544768	.744322
230	.0155564	8.64855	1446.58	6173.25	.0412359	.785183
235	.0170832	7.85165	1411.5	7155.03	.0372995	.822776
240	.0183716	7.11408	1375.15	8122.15	.0312339	.853
245	.0197565	6.44373	1339.12	9052.37	.0295405	.884341
250	.0212017	5.84162	1304.82	9967.22	.0215581	.906437
255	.0227899	5.30575	1271.49	10805.8	.0175594	.924035
260	.0244323	4.83319	1240.32	11583	.0148472	.938843
265	.0261169	4.42737	1210.35	12445.5	.0122539	.951097
270	.0278579	4.03563	1181.56	13185.1	.0101531	.96125
275	.0295568	3.71223	1157.13	13876.2	8.35739E-03	.972047
280	.0314715	3.41682	1132.55	14535.9	6.84354E-03	.976591
285	.0332566	3.17655	1111.68	15125.5	5.50737E-03	.982458
290	.0351133	2.95761	1091.54	15682.3	4.45871E-03	.986549
295	.0369126	2.75735	1073.54	16222.8	3.53335E-03	.990533
300	.038659	2.5812	1057.06	16672.3	2.81759E-03	.99335
305	.0403674	2.45252	1042.11	17121.8	2.17324E-03	.995523
310	.0419321	2.32207	1029.07	17492.3	1.63342E-03	.997157
315	.0435157	2.22446	1017.33	17835.8	1.18332E-03	.99834
320	.0449355	2.13208	1006.66	18146.1	8.12775E-04	.999152
325	.0461335	2.05485	998.074	18418.9	5.11133E-04	.999663
330	.0473372	1.99973	990.653	18652.2	2.76311E-04	.999939
335	.0482372	1.93519	984.524	18842.5	6.23179E-05	1
			COMBUSTION COMPLETED			
340	.0492219	1.8932	979.252	19003.3	0	1
345	.0498816	1.86039	975.179	19132.1	0	1
350	.0503384	1.83729	972.276	19222.3	0	1
355	.0506468	1.82365	970.536	19272.1	0	1
360	.0507433	1.81931	969.957	19290.7	0	1

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.75828 BARS
POWER (4 STROKE) = 108.765 KILOWATTS
SPECIFIC FUEL CONSUMPTION = 2.04[illegible]E-03 KG/KW-HR
THERMAL EFFICIENCY = 41.9982 PERCENT

Figure 16

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101487	10.1325	1090	0	0	0
	FUEL INJECTION START AT 185		COMBUSTION COMMENCED			
185	.0102065	10.3717	1121.73	55.4263	.0374117	.0374117
190	.0103805	10.9152	1193.87	244.477	.071854	.109266
195	.0106722	11.5212	1307.42	572.237	.102853	.212119
200	.0110754	11.2037	1428.41	1052.35	.129254	.341373
205	.011596	11.8167	1495.59	1587.83	.112392	.453544
	FUEL INJECTION STOP AT 205					
210	.0122312	11.8242	1523.14	2453.81	.0938122	.547357
215	.0129824	11.1335	1522.54	3310.21	.0766873	.624045
220	.0138422	10.3312	1501.78	4234.95	.0655108	.689555
225	.014815	9.484	1473.73	5138.74	.0544762	.744032
230	.0158934	8.64555	1446.98	6173.05	.0452559	.789288
235	.0170831	7.85105	1411.5	7153.09	.0378883	.827776
240	.0183713	7.11436	1375.15	8122.15	.0312238	.859
245	.0197562	6.44373	1335.12	9068.57	.0259405	.884941
250	.0212217	5.84262	1304.62	9957.22	.0213561	.906297
255	.0227593	5.30576	1271.49	10825.8	.0176934	.923991
260	.0244223	4.82919	1240.02	11663	.0148472	.938838
265	.0261183	4.40737	1210.36	12446.5	.0122939	.951132
270	.0278573	4.03683	1182.58	13185.1	.0101531	.961285
275	.0296558	3.71209	1157.19	13878.2	8.357392E-03	.969642
280	.0314715	3.42552	1133.55	14525.9	6.843546E-03	.976485
285	.0333365	3.17665	1111.66	15128.5	5.567375E-03	.982053
290	.0351159	2.95761	1091.34	15686.5	4.450715E-03	.986503
295	.0369126	2.75735	1072.54	16200.6	3.563365E-03	.990067
300	.038669	2.5812	1057.28	16672.3	2.617532E-03	.992685
305	.0403574	2.45652	1041.11	17101.6	2.170042E-03	.994855
310	.0419921	2.33207	1025.07	17492.3	1.633422E-03	.997157
315	.0435157	2.22446	1017.33	17839.8	1.183322E-03	.99834
320	.0449335	2.13226	1006.83	18148.1	8.151772E-04	.999152
325	.0462338	2.05418	999.074	18418.3	5.111332E-04	.999663
330	.0473372	1.99979	990.853	18652.2	2.782115E-04	.999939
335	.0483872	1.93819	984.524	18846.5	6.251752E-05	1
			COMBUSTION COMPLETED			
340	.0492219	1.8932	978.252	19008.3	0	1
345	.0499816	1.86239	975.179	19132.1	0	1
350	.0503584	1.83735	972.276	19228.3	0	1
355	.0505413	1.82356	970.536	19272.1	0	1
360	.0507423	1.81921	969.357	19290.7	0	1

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.75323 BARS
POWER (4 STROKE) = 195.755 KILOWATTS
SPECIFIC FUEL CONSUMPTION = 2.04413E-03 KG/KW-HR
THERMAL EFFICIENCY = 41.9322 PERCENT

Figure 17

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M^3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101467	12.1035	1090	0	0	0
185	.0102255	12.0213	1094.35	51.4157	0	0
FUEL INJECTION START AT 190			COMBUSTION COMMENCED			
190	.0103945	12.135	1113.71	[illegible]	.037071	.037072
195	.0105702	[illegible]	1137.76	502.443	.0702525	[illegible]
200	.010754	11.8025	1251.44	[illegible]	[illegible]	[illegible]
205	.011596	11.5021	1403.41	[illegible]	.155759	[illegible]
210	.0113313	11.4233	1478.14	[illegible]	.12338-	.444237
FUEL INJECTION STOP AT 212						
215	.0116384	10.5386	1524.63	3123	.0322352	.533353
220	.0108482	10.331	1527.73	4241.69	.0775564	.614451
225	.01-815	9.57022	1493.39	5323.66	.0549489	.673332
230	.0158354	9.78112	1472.21	6231.81	.021484	.730276
235	.0173522	8.01237	1442.78	3335.37	.0455511	.773007
240	.0162716	7.33451	1408.54	7331.63	.0270158	.818357
245	.0137565	6.81533	1375.34	8345.21	.021721	.842273
250	.0212317	6.02557	1342.12	9377.51	.0255338	.878231
255	.0227693	5.46323	1312.63	12771.7	.0222453	.857477
260	.0244223	4.3851	1282.31	11584.5	.0166344	[illegible]
265	.0261183	4.55337	1251.45	12424.3	.0153339	[illegible]
270	.0278575	4.17885	1224.31	13156.3	.0136511	.944786
275	.0296566	3.64355	1198.33	133.5	.0121574	.955333
280	.0314715	3.55421	1175.61	14537.4	5.077685E-03	.954741
285	.0332956	3.2364	1154.36	15212.5	7.331855E-03	.972273
290	.0351159	3.07418	1134.64	15752.5	5.217385E-03	.976-5
295	.0369125	2.87325	1116.35	15327.2	5.85384E-03	.983338
300	.038669	2.72735	1100.53	16817.7	4.145255E-03	.987764
305	.0403574	2.55531	1085.57	17335.1	3.331455E-03	.991085
310	.0419981	2.42121	1072.65	17672	2.657855E-03	[illegible]
315	.0435157	2.33112	1061.55	18333.5	2.2-5445E-03	.995731
320	.0449335	2.22253	1051.53	18555.3	1.542512E-03	.9973
325	.0462335	2.14727	1043.17	18835.3	1.127875E-03	.998417
330	.0473872	2.06015	1035.83	18883.1	7.752812E-04	.999223
335	.0483572	2.02537	1029.55	19238.4	4.93355E-04	.999333
340	.0492219	1.98147	1024.9	19255.7	2.555685E-04	.999932
345	.0498816	1.94735	1022.75	19335.2	4.735-5E-05	1
			COMBUSTION COMPLETED			
350	.0503354	1.93383	1017.74	19477.5	0	1
355	.0506468	1.92333	1015.93	19532.6	0	1
360	.0507433	1.92413	1015.33	19551.2	0	1

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.8163 BARS
POWER (4 STROKE) = 155.422 KILOWATTS
SPECIFIC FUEL CONSUMPTION = 2.0[illegible]E-03 KG/KW-HR
THERMAL EFFICIENCY = 41.4905 PERCENT

Figure 18

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.2121487	12.1225	1092	0	0	0
185	.2122256	12.2123	1094.35	33.4157	0	0
190	.2123605	5.71566	1074.27	232.45	0	0
FUEL INJECTION START AT 195			COMBUSTION COMMENCED			
195	.2125702	5.73482	1122.51	512.697	.020434	.020434
200	.2127754	9.32535	1171.34	912.495	.0574231	.105327
205	.212592	10.3642	1271.55	1441.15	.0552463	.162331
210	.2122302	12.7213	1337.22	2112.24	.122373	.283707
215	.2123824	12.6117	1454.52	2311.72	.12225	.405333
FUEL INJECTION STOP AT 215						
220	.2133483	12.1703	1455.49	3307.27	.0902348	.495562
225	.214315	9.34816	1431.56	4766.33	.0752732	.570835
230	.2153364	9.5043	1451.49	5722.8	.0644224	.637874
235	.2172022	8.12173	1451.43	6757.57	.0541841	.712232
240	.2183715	7.42217	1435.72	7752.91	.0455283	.733817
245	.2197358	6.78835	1427.12	8731.39	.038837	.822154
250	.2212317	6.15457	1377.32	9725.16	.0322925	.827449
255	.2227839	5.62219	1347.53	10523.3	.0272861	.854885
260	.2244223	5.1343	1316.95	11521.1	.0223567	.857822
265	.2261188	4.72154	1291.46	12335.4	.0154237	.927126
270	.2278579	4.31804	1265.37	13124.2	.0184672	.923510
275	.2295368	3.97343	1242.66	13858.4	.0132237	.937366
280	.2314715	3.65123	1216.21	14551.5	.0118627	.949843
285	.2332366	3.42827	1197.31	15225.6	9.12883E-03	.959366
290	.2351159	3.19197	1176.23	15811.1	8.25553E-03	.967142
295	.2369128	2.99174	1160.73	16366.2	6.52237E-03	.974045
300	.238839	2.8123	1144.85	16871.7	5.74221E-03	.979758
305	.2403574	2.6657	1132.95	17342.3	4.74128E-03	.984527
310	.2413301	2.53444	1116.45	17764.2	3.67347E-03	.988485
315	.2425157	2.42114	1127.35	18143.2	3.13662E-03	.991343
320	.2443395	2.32485	1257.97	18462.1	2.43743E-03	.99404
325	.2462226	2.243	1288.6	18775.7	1.94716E-03	.995966
330	.2473872	2.17225	1232.14	19030.5	1.47235E-03	.997424
335	.2483572	2.11382	1277.86	19245.3	1.22222E-03	.99855
340	.2492219	2.07319	1273.33	19428.4	7.42253E-04	.99925
345	.2498516	2.04274	1265.64	19556.1	4.55736E-04	.999745
350	.2503384	2.21619	1233.66	19622.3	2.23157E-04	.999974
355	.2505466	2.22126	1265.03	19712.3	2.57432E-05	1
			COMBUSTION COMPLETED			
360	.2507433	1.99683	1264.4	19732.1	0	1

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.35288 BARS
POWER (- STROKE) = 221.247 KILOWATTS
SPECIFIC FUEL CONSUMPTION = 1.99999E-03 KG/kW-HR
THERMAL EFFICIENCY = 42.8870 PERCENT

Figure 19

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101487	10.1325	1090	0	0	0
185	.0102255	10.0335	1084.36	53.4157	0	0
190	.0103825	9.71066	1074.67	203.45	0	0
195	.0106722	9.10233	1061.24	527.321	0	0
FUEL INJECTION START AT 200			COMBUSTION COMMENCED			
200	.0110754	8.34011	1034.43	955.16	.0357407	.0357407
205	.011596	8.00221	1151.54	1363.73	.0678573	.103598
210	.0122312	8.03919	1252.11	1974.54	.0964099	.200008
215	.0129824	8.54166	1364.27	2728.75	.122791	.32279
220	.0138422	8.75725	1432.19	3555.34	.184335	.407114
FUEL INJECTION STOP AT 220						
225	.014815	9.37574	1465.9	4431.34	.0091063	.51463
230	.0159554	8.88895	1475.58	5474.3	.0758015	.590561
235	.0172631	8.10353	1455.66	6451.19	.054014	.653575
240	.0186716	7.57158	1452.51	7451.41	.05414	.708016
245	.0197593	6.88341	1422.09	8438.19	.0453103	.753327
250	.0212317	6.2551	1427.25	9422.32	.038823	.792232
255	.0227339	5.75926	1391.25	10355.4	.032316	.825548
260	.0244023	5.27978	1354.66	11255.7	.027323	.851667
265	.0260133	4.83733	1335.33	12157.2	.0237327	.87723
270	.0276673	4.42172	1304.64	12959.7	.0203455	.897522
275	.0293383	4.10946	1281.65	13735.5	.0173413	.914777
280	.0314715	3.82731	1255.91	14453.3	.0146821	.929455
285	.0331755	3.54136	1233.74	15114.4	.0124545	.941954
290	.0351159	3.32287	1221.6	15747.5	.0108215	.952578
295	.0369126	3.10547	1205	16313.8	9.07222E-03	.961832
300	.0385553	2.93756	1189.95	16823.6	7.61864E-03	.969135
305	.0401674	2.77381	1176.91	17337.7	6.43598E-03	.975385
310	.0418321	2.64033	1165.25	17777	5.38357E-03	.98057
315	.0433137	2.52515	1154.97	18178.1	4.45381E-03	.985424
320	.0446332	2.42745	1146.46	18533.7	3.65288E-03	.989087
325	.0461336	2.3445	1139.15	18863.4	2.97323E-03	.99205
330	.0470372	2.27564	1132.36	19215.9	2.37383E-03	.994434
335	.0483372	2.22033	1128.38	19383.7	1.87763E-03	.996324
340	.0492215	2.17522	1125.23	19557.2	1.42645E-03	.997713
345	.0498216	2.14115	1122.19	19649.6	1.01333E-03	.998727
350	.0502564	2.11674	1119.97	19751.1	6.81433E-04	.999408
355	.0506456	2.10221	1118.63	19812	4.06673E-04	.999815
360	.0507433	2.09848	1118.77	19832.2	1.85072E-04	1
			COMBUSTION COMPLETED			

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.88543 BARS
POWER (4 STROKE) = 332.689 KILOWATTS
SPECIFIC FUEL CONSUMPTION = 1.88886E-03 KG/KW-HR
THERMAL EFFICIENCY = 42.1832 PERCENT

Figure 20

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101487	10.1325	1090	0	0	0
185	.0102238	10.0033	1084.35	53.4157	0	0
190	.0104823	[illegible]	1074.57	[illegible]	0	0
195	.0108722	9.38195	1061.84	507.601	0	0
200	.0113734	8.85717	1044.51	671.811	0	0
FUEL INJECTION START AT 200	COMBUSTION COMMENCED					
205	.011998	8.67[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
210	.0123312	8.72845	1132.37	1885.24	[illegible]	[illegible]
215	.0125824	8.51754	1226.85	2349.32	[illegible]	[illegible]
220	.0138422	9.15[illegible]	1322.73	3336.34	.11753	[illegible]
225	.014615	9.00174	1408.65	4211.31	.121558	.414101
FUEL INJECTION STOP AT 225						
230	.0153924	8.81725	1445.53	5154.81	.0373233	[illegible]
235	.0172832	8.18128	1455.83	6155.18	.0747[illegible]	[illegible]
240	.0190375	7.53547	1456.12	7133.79	[illegible]	[illegible]
245	.0207565	6.35013	1446.71	8155.72	.0541388	[illegible]
250	.0218317	6.3313	1423.3	9145.54	.0461233	.740164
255	.0227699	5.67231	1428.52	10125.2	[illegible]	[illegible]
260	.0244123	5.19249	1362.16	[illegible]	.0305782	[illegible]
265	.0261189	4.36238	1353.87	[illegible]	.0267211	[illegible]
270	.0278579	4.5757	1341.82	13737	.0146228	.888387
275	.0295298	4.10314	1312.56	13515	.0210941	[illegible]
280	.0314715	3.9199	1300.42	14225.6	.0121232	[illegible]
285	.0332556	3.6622	1281.5	14958.2	.0155348	[illegible]
290	.0351159	3.42537	1264.73	15522.7	.0130029	.964449
295	.0369125	3.21917	1248.89	16155.6	.0114132	[illegible]
300	.0386569	3.03663	1235.32	16749.2	9.76006E-03	[illegible]
305	.0403674	2.88175	1222.83	17252	8.34855E-03	[illegible]
310	.0419901	2.74553	1212.27	17726.6	7.03352E-03	.971125
315	.0435197	2.62808	1203.26	18113.9	5.86737E-03	[illegible]
320	.0449395	2.53241	1195.17	18465.2	[illegible]	[illegible]
325	.0462336	2.44885	1188.34	18805.2	4.21011E-03	[illegible]
330	.0473872	2.37795	1184.27	19085.6	3.47287E-03	.989845
335	.0483872	2.32143	1180.48	19321.5	2.82195E-03	[illegible]
340	.0492219	2.27725	1177.73	19513.4	2.25764E-03	[illegible]
345	.0498816	2.24328	1175.64	19662.5	1.75563E-03	[illegible]
350	.0503584	2.21928	1174.22	19763.9	1.31345E-03	[illegible]
355	.0506463	2.20548	1174.12	19833.7	9.32653E-04	[illegible]
357	.0507433	2.20085	1174.97	19854	9.11895E-04	[illegible]

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.8508 BARS
POWER (4 STROKE) = 226.511 KILOWATTS
SPECIFIC FUEL CONSUMPTION = [illegible] KG/KW-HR
THERMAL EFFICIENCY = [illegible] PERCENT

Figure 21

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101467	12.1325	1292	0	0	0
185	.0103035	12.0102	1284.35	53.4137	0	0
190	.0105568	9.75666	1274.67	130.45	0	0
195	[illegible]	[illegible]	1261.24	237.381	0	0
FUEL INJECTION START AT 200			COMBUSTION COMPLETED			
200	[illegible]	9.73372	1142.57	355.223	.0719655	.0719655
205	.0114585	12.5616	1310.47	1422.74	.141432	.214331
210	[illegible]	13.3731	1418.45	[illegible]	.133376	.333376
FUEL INJECTION START AT 210						
215	.0123324	10.7333	1478.37	3323.62	.125178	.444154
220	.0133433	10.2235	1520.3	3317.33	.2933-17	.534233
225	.014313	9.33243	1524.64	4327.13	.0757571	.625301
230	.0153324	8.58244	1490.35	5223.32	.0536773	.670161
235	.0170832	8.15371	1483.23	6221.53	.0324533	.727213
240	.0180716	7.44555	1442.51	7837.01	.0448355	.771515
245	.0197535	6.73220	1413.33	8311.33	.0377354	.809334
250	.0212317	6.17339	1379.35	9767.33	.0317625	.841415
255	.0227333	5.62289	1346.67	10237.2	.0237566	.868173
260	.0244223	5.1361	1313.38	11555.5	.022571	.890744
265	.0261183	4.70116	1291.23	11393.9	.0182442	.908733
270	.0275873	4.31612	1264.73	12126.6	.0150567	.925657
275	.0286223	3.97342	1239.3	12932.3	.0128385	.935436
280	.0214715	3.67753	1216.74	14634.7	.0114126	.95032
285	.0222355	3.41534	1195.73	15272.1	9.22227E-03	.962415
290	.0231155	3.16735	1176.47	15572.8	6.24243E-03	.968456
295	.0239125	2.98574	1156.8	16437.4	6.72529E-03	.975185
300	.025563	2.8117	1141.74	16336.6	5.55395E-03	.980732
305	.0423374	2.66334	1123.67	17421.3	4.33343E-03	.985318
310	.0413321	2.52393	1116.64	17633.4	3.7414E-03	.98906
315	.2433197	2.41359	1104.8	13322.5	3.01322E-03	.992074
320	.2443358	2.31722	1093.31	18536.7	2.32662E-03	.994403
325	.2468333	2.13731	1087.24	13331.3	1.35173E-03	.996314
330	.2473573	2.1633	1080.25	13265.7	1.33331E-03	.997729
335	.2483572	2.114	1074.9	13233.8	1.21432E-03	.998721
340	.2492219	2.06301	1072.3	13474.4	6.73735E-04	.999432
345	.0493315	2.03431	1065.54	13323.8	4.25236E-04	.999822
350	.2502534	2.01221	1063.71	13708.2	1.83042E-04	.999937
355	.2525453	1.99535	1061.64	13734	2.31142E-05	1
			COMBUSTION COMPLETED			
360	.0507433	1.9923	1061.32	13732.2	0	1

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.87025 BARS
[illegible]
[illegible]
[illegible] EFFICIENCY = [illegible] PERCENT

Figure 22

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M↑2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M↑3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL IN STEP (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101487	10.1325	1090	0	0	0
185	.0102835	10.0319	1084.38	50.4157	0	0
190	.0105305	9.72222	1074.57	200.45	0	0
195	.0109702	9.38132	1061.34	507.601	0	0
200	.0114754	8.85717	1044.51	876.211	0	0
FUEL INJECTION START AT 205			COMBUSTION COMMENCED			
205	.012535	9.12745	1122.43	1247.59	.0711304	.0711304
210	.0128818	9.9573	1220.87	1851.45	.137254	.208754
215	.0135604	10.155	1331.2	2786.11	.122551	.331132
FUEL INJECTION STOP AT 215						
220	.0143422	9.5523	1454.89	3371.6	.10385	.435135
225	.014815	9.45776	1493.15	4318.19	.0864675	.521782
230	.0159554	8.88043	1488.59	5511.35	.0749275	.5957
235	.0172232	8.21601	1473.24	6605.91	.0625255	.658235
240	.018317.5	7.54328	1450.8	7541.44	.0534558	.712515
245	.0197555	6.90735	1427.14	8542.44	.0458236	.758725
250	.0212317	6.31229	1418.81	9517.53	.0382588	.796984
255	.0227833	5.78234	1381.47	10456.6	.0324112	.829427
260	.0244223	5.27727	1356.19	11330.3	.0275162	.856903
265	.0261138	4.84125	1332.1	12215.7	.0231653	.880068
270	.0278579	4.45227	1305.83	13021.3	.019883	.90015
275	.0296568	4.12833	1281.44	13707.3	.0159814	.91712
280	.0314715	3.83257	1258.32	14315.4	.0142348	.931354
285	.0332926	3.5789	1232.82	15185.6	.0112132	.942571
290	.0351153	3.33568	1222.41	15686.2	.0102089	.95214
295	.0369126	3.12162	1201.37	16081.3	8.75782E-03	.960937
300	.038689	2.93075	1185.31	16513	7.46132E-03	.970353
305	.0403674	2.76330	1170.88	17356.5	6.24291E-03	.976843
310	.0419381	2.62533	1160.25	17533	5.17215E-03	.981813
315	.0433157	2.50247	1151.88	18113.4	4.31237E-03	.98613
320	.0446335	2.40263	1144.19	18550.3	3.33332E-03	.989463
325	.0462335	2.32224	1135.74	18916.4	2.65352E-03	.992152
330	.0472872	2.27276	1132.63	19154.3	2.27682E-03	.994485
335	.0483872	2.21517	1125.33	19378.6	1.72274E-03	.996533
340	.0492219	2.17029	1122.48	19561.7	1.22283E-03	.997965
345	.0499316	2.13572	1118.35	19703.7	5.45042E-04	.998875
350	.0503584	2.11121	1117.25	19825	6.27015E-04	.999502
355	.0506468	2.0968	1115.65	19885.6	3.83556E-04	.999884
360	.0507423	2.09273	1115.72	19935.8	1.35556E-04	1
			COMBUSTION COMPLETED			

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.69854 BARS
POWER (4 STROKE) = 201.935 KILOWATTS
SPECIFIC FUEL CONSUMPTION = .9515E-03 KG/KW-HR
THERMAL EFFICIENCY = 45.2258 PERCENT

Figure 23

DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M^2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

COMPRESSION ANGLE (DEGREES)	CYLINDER VOLUME (M^3)	CYLINDER PRESSURE (BAR)	CYLINDER TEMPERATURE (DEG K)	CYLINDER WORK (JOULES)	FUEL INJECTED (FRACTION)	CUMULATIVE FUEL (FRACTION)
180	.0101487	10.1325	1090	0	0	0
185	.0103056	10.0228	1084.38	51.4157	0	0
190	.0106625	9.76628	1074.67	230.45	0	0
195	.0109702	9.33095	1061.14	527.891	0	0
200	.0114754	8.85717	1044.31	878.811	0	0
205	.0119595	8.33936	1025.22	1282.21	0	0
FUEL INJECTION START AT 210			COMBUSTION COMMENCED			
210	.0123313	8.43314	1035.54	1681.34	.0080082	.0080081
215	.0128324	9.17124	1155.63	2522.68	.110571	.10864
220	.0133422	9.23974	1265.4	3320.49	.117484	.320128
FUEL INJECTION STOP AT 230						
225	.014415	9.14285	1432.36	4213.37	.121576	.421879
230	.0158554	8.71881	1455.47	5085.05	.0887614	.508651
235	.0172932	8.17235	1472.45	5867.41	.0741053	.582737
240	.0183716	7.57387	1467.81	7222.2	.0600051	.643772
245	.0197353	6.95536	1453.93	8810.76	.0502546	.693308
250	.0212217	6.41632	1434.7	9183.23	.0455538	.744866
255	.0227899	5.93728	1410.43	10157.5	.0353124	.782539
260	.0244003	5.42328	1385.59	11073.4	.0301173	.818816
265	.0261188	4.93358	1365.81	11958.3	.032011	.847937
270	.0278573	4.58103	1343.86	12734.5	.0242003	.873027
275	.0295533	4.20545	1321.22	13583.1	.0227537	.896117
280	.0314715	3.92355	1300.6	14084	.0177381	.907318
285	.0332955	3.65377	1281.39	15016.5	.0151118	.922177
290	.0351159	3.49397	1264.2	15592.9	.010865	.936863
295	.0369125	3.21709	1248.47	16057.5	.0112057	.947488
300	.038553	3.02523	1234.24	16606.7	9.5776E-03	.957046
305	.0403574	2.87874	1221.54	17309	8.18398E-03	.965289
310	.0419321	2.74328	1210.78	17765.1	6.91002E-03	.972143
315	.0435197	2.62645	1201.39	18172.8	5.83622E-03	.977993
320	.0449335	2.52655	1193.34	18541.6	4.91237E-03	.982913
325	.0462336	2.44273	1186.99	18861.1	4.09715E-03	.987
330	.0472872	2.37371	1182.14	19141	3.38184E-03	.990382
335	.0481873	2.31731	1178.23	19375.5	2.78228E-03	.993064
340	.0488219	2.27246	1175.39	19567	2.1726E-03	.995183
345	.0493816	2.23834	1173.15	19715.8	1.67828E-03	.996943
350	.0500384	2.2144	1171.64	19828	1.24522E-03	.998187
355	.0506488	2.20147	1171.47	19885.6	9.72375E-04	.999039
360	.0507433	2.19972	1172.33	19896.9	5.62866E-04	.999522

EXHAUST VALVE OPEN - - CYCLE COMPLETE

IMEP = 4.90222 BARS
POWER (4 STROKE) = 823.85 KILOWATTS
SPECIFIC FUEL CONSUMPTION = 1.98535E-03 KG/KW-HR
THERMAL EFFICIENCY = -3.3715 PERCENT

Figure 24

Appendix A

Specifications of Test Remley Engine[22]

Type of Engine	Four Stroke
Bore	4.0 inches
Stroke	2.5 inches
Cylinder Displacement	31.41 cubic inches
Connecting Rod Length	6.25 inches
Compression Ratio	14.3 : 1
Number of Compression Rings	2
Number of Oil Rings	1
Number of Inlet Valves	2
Number of Exhaust Valves	2
Valve Diameter	1.286 inches
Valve Lift	0.280 inches
Inlet Valve Timing open/close	15^{o}BTDC/50^{o}ABDC
Exhaust Valve Timing open/close	50^{o}BBDC/15^{o}ATDC
Diameter of Intake Manifold Pipe	2.00 inches
Diameter of Exhaust Manifold Pipe	1.60 inches

Data Collected from Test Run[22]

Engine Speed in RPM	1450
Inlet Pressure	13.5 inches Hg gage
Exhaust Pressure	13.4 inches Hg gage
Inlet Air Temperature	186° F
Air to Fuel Ratio	25.38
IMEP	88.1 psi
Start of Injection	12.5° BTDC
Ignition Delay	5.5°
Period of Fuel Injection	17.5°

Appendix B

Computer Model

The computer program is written in TRS-80 Model III Disk Basic and consists of a main program and nine subroutines. The program listing has numerous remarks statements inserted to make the algorithm and computer code easier to understand. Since the program takes a considerable length of time to run, it is recommended that the remark statements be deleted before running. Samples of output are presented in figures 15 to 24.

```
5 '*******************************************************
6 '**********           MAIN PROGRAM           **********
7 '*******************************************************
8 'This program is written in TRS-80 Model III Disk Basic.
9 'Remove all remarks spaces before running to speed up run time.
10 'Dimension arrays.
11 'Array U contains the thermodynamic polynomial coefficients.
12 'Array F(5) and FD(5) are used in subroutines for calculating
13 'Thermodynamic data (i.e. enthalpy, internal energy, and moles).
14 'Arrays A(5) and B(5) contain number of moles of the
15 'Five species at the beginning of step,  A,  and at
16 'The end of the step, B.
20 DIM U(5,5), F(5), FD(5), A(5), B(5)
25 'Define all variables starting with I & J as integers.
30 DEFINT I,J
35 'Input data is requested from the operator -- WATCH UNITS
40 INPUT"ENTER TODAY'S DATE";DATE$
50 INPUT"ENTER RUN NUMBER";NUMB$
60 INPUT"ENTER CYLINDER BORE IN METERS";D
70 INPUT"ENTER STROKE IN METERS";S
80 INPUT"ENTER CONNECTING ROD LENGTH IN METERS";L
90 INPUT"ENTER ENGINE SPEED IN RPM";RPM
100 INPUT"ENTER ENGINE COMPRESSION RATIO";CR
110 INPUT"ENTER AIR / FUEL RATIO";AFR
120 INPUT"ENTER PRESSURE AT START OF COMPRESSION IN N/M↑2";P1
130 INPUT"ENTER TEMPERATURE AT START OF COMPRESSION IN DEG K";T1
140 INPUT"ENTER RESIDUAL GAS FRACTION";F
150 INPUT"ENTER CRANK ANGLE FOR INTAKE VALVE SHUT";ALPHA
160 INPUT"ENTER CRANK ANGLE FOR EXHAUST VALVE OPEN";AEVO
170 INPUT"ENTER CRANK ANGLE FOR FUEL INJECTION ";AIJECT
```

```
180 INPUT"ENTER PERIOD OF FUEL INJECTION (DEGREES)";IJECT
185 INPUT"ENTER CRANK ANGLE INCREMENTS FOR THIS RUN";ADELT
190 INPUT"SELECT FUEL:  (1) FOR C8H18 (OCTANE)    (2) FOR C3H8  (PROPANE)";NN
195 'Load in fuel data for selected fuel.
200 IF NN=1 THEN GOSUB 4000 ELSE GOSUB 4100
205 'Load in constants with subroutine 4200
210 GOSUB 4200
215 'Subroutine 5000 calculates cylinder volume and surface area.
220 GOSUB 5000
225 V1=V
227 'Calculate phi and number of moles of fuel based on perfect combustion
230 GOSUB 5100
235 'Print out input data.
240 GOSUB 5200
250 PRES=P1*PBAR
255 IF AIJECT=ALPHA THEN LPRINT,,"FUEL INJECTION START AT ";AIJECT;"COMBUSTION COMMENCED"
260 LPRINT,ALPHA,V1,PRES,T1,WRKT,DMF,KO
265 'Convert heat of reaction from J/Kg to J/Kgmole.
270 QVS=QVS*WF
275 'Calculate moles of all species at beginning of run.
280 A(1)=MOLE*CA*F               'Moles of Carbon dioxide
290 A(2)=MOLE*F*HA/2             'Moles of Water Vapor
300 A(3)=MOLE*SOX*(1-F)/PHI      'Moles of Oxygen
310 A(4)=3.76*A(3)               'Moles of Nitrogen
320 A(5)=0.0                     'Moles of Fuel
330 FORII=1TO5
340 B(II)=A(II):X(II)=A(II)
350 NEXTII
355 'Y, I1, & I2 are values that are fed into subroutines 5500 & 6000
356 'These subroutines are used to calculate internal energy, enthalpy and moles.
```

```
357 'The temperatures are raised to powers in the polynomial expressions.
358 'The values of I1 & I2 tell the subroutines for what species to solve.
360 Y=TS:I1=1:I2=4
370 FOR I=I1 TO I2
380 GOSUB 5500
390 NEXTI
400 GOSUB 6000
405 'Calculate internal energy, E, at reference temperature, TS.
410 ES1=RMOL*TS*F1
420 Y=T1
430 FOR I=I1 TO I2
440 GOSUB 5500
450 NEXT I
460 GOSUB 6000
470 M1=F3
475 'Calculate internal energy and Specific heat at temperature T1
480 E1=RMOL*T1*F1
490 C1V=RMOL*F2/F3
495 'Add crank angle interval to go to end of step
500 ALPHA=ALPHA+ADELT
505 'Calculate cylinder volume and area.
510 GOSUB 5000
520 V2=V
525 IF KO >= 1.0 GOTO 540
530 IF ALPHA >= AIJECT THEN GOTO 560
540 DMF=0.0
550 GOTO 590
560 KK=1
565 'If injection has occurred, then go to Combustion Subroutine.
```

```
570 GOSUB 6500
580 IF KK = 2 GOTO 600
585 'First approximation at temperature.
590 T2=T1*((V1/V2)↑(RMOL/C1V))-(DMF*QVS*MOLES)/(C1V*M1)
595 'Calculate the internal energy after combustion at TS.
600 Y=TS:I2=4
610 FOR II=1 TO 5
620 X(II)=B(II)
630 NEXT II
640 FOR I=I1 TO I2
650 GOSUB 5500
660 NEXT I
670 GOSUB 6000
680 EES2=RMOL*TS*F1
685 'Calculate the internal energy & specific heat at T2.
690 Y=T2
700 FOR I=I1 TO I2
710 GOSUB 5500
720 NEXT I
730 GOSUB 6000
740 E2=RMOL*T2*F1
750 M2=F3
760 C2V=RMOL*F2/F3
765 'Calculate pressure at end of step - Ideal gas.
770 P2=(M2/M1)*(T2/T1)*(V1/V2)*P1
775 'Calculate heat transfer in subroutine 7000
780 GOSUB 7000
785 'Calculate work
790 DW=0.5*(P1+P2)*(V2-V1)
795 'Calculate error for Newton-Raphson iteration.
```

```
800 FE=(E2-EES2)-(E1-ES1)+DW-DQ+(DMF*MOLE*QVS)
810 EARER=FE/(M2*C2V)
815 'NRACC is the allowable error for Newton-Raphson Iteration.
820 IFABS(EARER)<NRACCGOTO860
830 T2=T2-EARER/2
835 'Recalculate energies and specific heats at "new" temperature.
840 IF ALPHA < AIJECT GOTO 690
850 KK=2:GOSUB 6600
855 GOTO 600
856 'Convert pressure to bars.
860 PRES=P2*PBAR
865 'Cumulative work
870 WRKT=WRKT+DW
875 'Cumulative heat transfer.
880 Q=Q+DQ
882 'Cumulative heat release.
885 KO=KO+DMF
890 LPRINT,ALPHA,V2,PRES,T2,WRKT,DMF,KO
892 IF YZ=10. GOTO 900
895 IF ZZ=1 THEN LPRINT,,,,"COMBUSTION COMPLETED" ELSE GOTO 900
896 YZ=10.
900 IF ALPHA = AEVO GOTO 2000
905 'Shift end of step data to beginning of next step.
910 P1=P2
920 V1=V2
930 T1=T2
940 E1=E2
950 ES1=EES2
```

```
960 C1V=C2V
970 M1=M2
975 PEP=PPEP
976 RACT=RRCT
980 FOR II=1 TO 5
990 A(II)=B(II)
1000 NEXT II
1010 GOTO 500
2000 LPRINT,"EXHAUST VALVE OPEN - - CYCLE COMPLETE"
2010 'Calculate power, IMEP, efficiency, and sfc.
2020 PWR=WRKT*RPM*1.2E-05
2030 MEP=WRKT*PBAR/VS
2040 EFFTH=100.0*WRKT/(-QVS*MOLE)
2050 SFC=(3.6E06*WF)/(-QVS*KO*EFFTH)
2060 LPRINT:LPRINT
2070 LPRINT TAB(20)"IMEP =";MEP;"  BARS"
2080 LPRINT TAB(20)"POWER (4 STROKE) =";PWR;"  KILOWATTS"
2090 LPRINT TAB(20)"SPECIFIC FUEL COMSUMPTION =";SFC;"  KG/KW-HR"
2100 LPRINT TAB(20)"THERMAL EFFICIENCY =";EFFTH;"  PERCENT"
2110 END
```

```
4000 '**************************************************
4001 '**********            FUEL DATA           **********
4002 '**************************************************
4010 FUEL$="C8H18 (ISO OCTANE)"
4015 'Polynomial coefficients for iso-octane
4020 U(5,1)=-0.71993
4030 U(5,2)=4.6426E-02
4040 U(5,3)=-1.68385E-05
4050 U(5,4)=-2.67009E-09
4060 U(5,5)=0.0
4070 CA=8:HA=18            'Carbon atoms = 8, Hydrogen atoms = 18
4080 QVS=-4.2E07           'Lower heating value in J/Kg.
4090 RETURN
4100 'Polynomial coefficients for propane.
4110 FUEL$="C3H8 (PROPANE)"
4120 U(5,1)=1.13711
4130 U(5,2)=1.45532E-02
4140 U(5,3)=-2.95876E-06
4150 U(5,4)=0.0
4160 U(5,5)=0.0
4170 CA=3:HA=8             'Carbon atoms = 3, Hydrogen atoms = 8.
4180 QVS=-4.63E07          'Lower heating value in J/Kg.
4190 RETURN
```

```
4200 '**************************************************
4201 '********** Constants and other input data ********
4202 '**************************************************
4208 'This subroutine loads various constants and
4209 'polynomial coefficients for CO2, H2O, N2 and O2.
4210 PO=101325              'Reference pressure in N/M↑2
4220 TS=298                 'Reference temperature in degrees Kelvin.
4230 PI=3.1415927
4240 RD=180/PI              'Converts from degrees to radians
4250 RMOL=8314.3            'Newton-meter/(kg-mole)K
4260 WRKT=0.0:Q=0.0
4270 PBAR=1E-05             'Conversion from N/m↑2 to bars.
4271 K1=0.014               'Preparation rate coefficient 0.008-0.022.
4272 K2=2/3                 'Preparation rate coefficient x.
4273 K3=6.5E11              'Reaction rate coefficient K'.
4274 K4=1.5E4               'Reaction rate coefficient act.
4275 NRACC=1.0              'Newton-Raphson Accuracy.
4276 G1=0.26:G2=0.75:G3=3.88E-08   'Annand equation coefficients a,b,&c.
4277 TW=750:PR=0.7          'Wall temperature (assumed) and Prandtl Number.
4278 M(1)=24E-06:M(2)=20E-06:M(3)=32E-06:M(4)=29E-06 'viscosity of species.
4280 FOR II=1 TO 5
4290 A(II)=0.0:B(II)=0.0
4300 NEXT II
4310 'Thermodynamic data preparation  U(I,J)
4320 'I=Species   J=Coefficient
4330 'Species: 1=CO2;  2=H2O;  3=O2;  4=N2;  5=FUEL
4350 'Carbon Dioxide
4360 U(1,1)=3.0959
4370 U(1,2)=2.73114E-03
4380 U(1,3)=-7.88542E-07
```

```
4390 U(1,4)=8.66002E-11
4400 U(1,5)=0.0
4430 'Water Vapor
4440 U(2,1)=3.74292
4450 U(2,2)=5.65590E-04
4460 U(2,3)=4.95240E-08
4470 U(2,4)=-1.81802E-11
4480 U(2,5)=0.0
4510 'Oxygen
4520 U(3,1)=3.25304
4530 U(3,2)=6.52350E-04
4540 U(3,3)=-1.49524E-07
4550 U(3,4)=1.53897E-11
4560 U(3,5)=0.0
4590 'Nitrogen
4600 U(4,1)=3.34435
4610 U(4,2)=2.94260E-04
4620 U(4,3)=1.95300E-09
4630 U(4,4)=-6.57470E-12
4640 U(4,5)=0.0
4700 'Volume of cylinder at BDC
4710 VS=PI*S*(D/2.0)↑2
4715 'Volume at TDC.
4720 VC=VS/(CR-1)
4730 N=L/(S/2)
4735 'Total cylinder volume.
4740 VT=VS+VC
4745 'Areas in cylinder.
4750 AC=4*VC/D
4760 AS=S*PI*D
4770 AT=AS+AC
4780 RETURN
```

```
5000 **************************************************
5001 **********   VOLUME & AREA OF CYLINDER  **********
5002 **************************************************
5010 SWEEP=(1+N-(N↑2.0-(SIN(ALPHA/RD))↑2.0)↑0.5-COS(ALPHA/RD))
5020 V=VT-(VS/2)*SWEEP
5030 AREA=AT-(AS/2)*SWEEP
5040 RETURN
```

```
5100 '**************************************************
5101 '**********      FUEL CALCULATIONS       **********
5102 '**************************************************
5105 'Calculate Number moles of fuel.
5110 SOX=CA+HA/4
5120 WF=12.0*CA+HA
5125 'Calculate air fuel ratio - stoichiometric.
5130 ASTF=4.76*SOX*28.96/WF
5140 PHI=ASTF/AFR
5145 'Calculate mole of fuel for perfect combustion.
5150 MOLE=P1*V1*PHI/(4.76*SOX*RMOL*T1)
5160 RETURN
```

```
5200 '************************************************
5201 '**********        PRINT SUBROUTINE      **********
5202 '************************************************
5205 LPRINT CHR$(27);"6":LPRINT CHR$(27);CHR$(49);CHR$(30)
5210 LPRINT:LPRINTTAB(25)"DIESEL ENGINE COMBUSTION CYCLE":LPRINT
5220 LPRINT TAB(25);"RUN: ";NUMB$;"  ";DATE$
5230 LPRINT TAB(10)"INPUT DATA:"
5240 LPRINT TAB(20)"CYLINDER BORE = ";D;"METERS"
5250 LPRINT TAB(20)"STROKE = ";S;"METERS"
5260 LPRINT TAB(20)"CONNECTING ROD LENGTH = ";L;"METERS"
5270 LPRINT TAB(20)"ENGINE SPEED = ";RPM;"RPM"
5280 LPRINT TAB(20)"ENGINE COMPRESSION RATIO = ";CR
5290 LPRINT TAB(20)"AIR / FUEL RATIO = ";AFR
5300 LPRINT TAB(20)"TRAPPED PRESSURE = ";P1;"N/M↑2"
5310 LPRINT TAB(20)"TRAPPED TEMPERATURE = ";T1;"DEG KELVIN"
5320 LPRINT TAB(20)"RESIDUAL AIR FRACTION = ";F
5330 LPRINT TAB(20)"FUEL SELECTED FOR THIS ANALYSIS = ";FUEL$
5340 LPRINT TAB(20)"     WITH A LOWER HEATING VALUE = ";QVS;"JOULES/KG"
5350 LPRINT TAB(15)"STOICHIOMETRIC AIR / FUEL RATIO = ";ASTF
5360 LPRINT TAB(15)"FUEL / AIR EQUIVALENCE (PHI) = ";PHI
5400 LPRINT CHR$(27);CHR$(48);CHR$(29)
5405 LPRINT CHR$(27);"8"
5410 LPRINT,"COMPRESSION"," CYLINDER"," CYLINDER"," CYLINDER"," CYLINDER","   FUEL","CUMULATI
5420 LPRINT,"   ANGLE","  VOLUME"," PRESSURE","TEMPERATURE","   WORK"," IN STEP","   FUEL"
5430 LPRINT," (DEGREES)","   (M↑3)","   (BAR)","  (DEG K)"," (JOULES)","(FRACTION)","(FRACTI
5440 LPRINT,"--------","--------","--------","--------","--------","--------","--------"
5450 RETURN
```

```
5500 '**************************************************
5505 '****** THERMODYNAMIC PROPERTIES OF MIXTURES ******
5510 '**************************************************
5520 F=0.0
5530 FD=0.0
5550 FOR J=1 TO 5
5560 LET Z=J
5570 LET L=J-1
5580 F=F+U(I,J)*Y↑(Z-1.0)
5600 FD=FD+Z*U(I,J)*Y↑(Z-1.0)
5620 NEXT J
5625 F(I)=F
5626 FD(I)=FD
5630 RETURN


6000 '**************************************************
6005 '****** THERMODYNAMIC PROPERTIES OF MIXTURES ******
6006 '**************************************************
6010 F1=0.0
6020 F2=0.0
6030 F3=0.0
6040 FOR I=I1 TO I2
6050 F1=F1+X(I)*(F(I)-1.0)
6070 F2=F2+X(I)*(FD(I)-1.0)
6090 F3=F3+X(I)
6100 NEXT I
6120 RETURN
```

```
6500 '*****************************************************
6501 '**********     COMBUSTION SUBROUTINE     **********
6502 '*****************************************************
6506 IF ALPHA = AIJECT LPRINT,, "FUEL INJECTION START AT ";AIJECT,"COMBUSTION COMMENCED"
6507 IF ALPHA = (AIJECT+IJECT+ADELT) LPRINT,,"FUEL INJECTION STOP AT ";ALPHA-ADELT
6508 IF KO >= 1.0 GOTO 540
6510 FOR II=1 TO 5
6515 X(II)=A(II)
6520 NEXT II
6540 'Variable MIJECT is the total fuel injected to this point.
6545 IF MIJECT = (MOLE*WF) THEN GOTO 6600
6550 MIJECT = MIJECT+MOLE*WF*ADELT/IJECT
6590 'Calculate Partial Pressure of Oxygen
6600 PO2=P2*PBAR*A(3)/M1
6605 'Calculate the fuel that has been prepared in this step.
6610 MASUNB=MIJECT-PEP
6615 Z1=MIJECT↑(1-K2)
6616 Z2=MASUNB↑K2
6617 Z3=PO2↑0.4
6620 PN1=K1*Z1*Z2*Z3
6625 'Calculate the cumulative fuel Prepared kg.
6630 PPEP=PEP+(PN1*ADELT)
6635 'Check to see if reaction rate less than preparation
6640 IF PPEP <= RACT GOTO 6690
6650 COEF=K3*PO2/(RPM*SQR(TM))
6660 R1N=COEF*(PPEP-RACT)*EXP(-K4/TM)
6665 'Calculate cumulative fuel reacted kg.
6670 RRCT=RACT+(R1N*ADELT)
6680 IF RRCT < PPEP THEN DF=R1N*ADELT ELSE DF=PN1*ADELT:GOTO6691
```

```
6690 DF=PN1*ADELT
6691 DMF=DF/(WF*MOLE)
6692 IF (KO+DMF)>1.0 THEN DMF=(1.0-KO)
6693 IF (KO+DMF)=1.0 THEN ZZ=1
6700 'Calculate the amount of fuel burned in this step.
6701 A(5)=DMF*MOLE:Y=TS:I2=5
6702 FOR I=1 TO I2:GOSUB 5500:NEXT I:GOSUB 6000
6703 ES1=RMOL*TS*F1:Y=T1
6704 FOR I=1 TO I2:GOSUB 5500:NEXT I:GOSUB 6000
6705 M1=F3-A(5)
6706 E1=RMOL*T1*F1
6707 C1V=RMOL*F2/F3
6709 'Calculate moles of products after combustion
6710 B(1)=A(1)+(DMF*MOLE*CA)
6720 B(2)=A(2)+(DMF*MOLE*HA/2)
6730 B(3)=A(3)-(DMF*MOLE*SOX)
6740 B(4)=A(4)
6750 B(5)=0.0
6760 RETURN
```

```
7000 '*****************************************************
7010 '********** HEAT TRANSFER-ANNAND EQUATION *********
7020 '*****************************************************
7055 'Calculate piston speed
7060 VP=2*S*RPM/60
7065 'Calculate mean temperature
7070 TM=(T1+T2)/2
7110 'Calculate the mixture viscosity from the individual specie's viscosity.
7120 MU=X(1)*M(1)+X(2)*M(2)+X(3)*M(3)+X(4)*M(4)
7125 'Calculate Specific Heat Cp
7130 CP=C2V+RMOL/M2
7135 'Calculate Conductivity
7140 K=CP*MU/PR
7145 'Calculate density
7150 RO=P2*M2/(RMOL*T2)
7155 'Calculate Reynold's Number
7160 RE=RO*D*VP/MU
7165 'Convective Term from Annand's Equation
7170 CVECT=G1*K*(RE↑G2)*(TM-TW)/D
7175 'Radiation Term from Annand's Equation
7180 CRAD=G3*(TM↑4.0-TW↑4.0)
7190 IF ALPHA < AIJECT THEN QDT=CVECT ELSE QDT=CVECT+CRAD
7195 'Calculate heat loss this iteration
7200 DQ=AREA*QDT*ADELT/(6*RPM)
7210 RETURN
```